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FINS OF THE FUTURE - FFG-7, (U)
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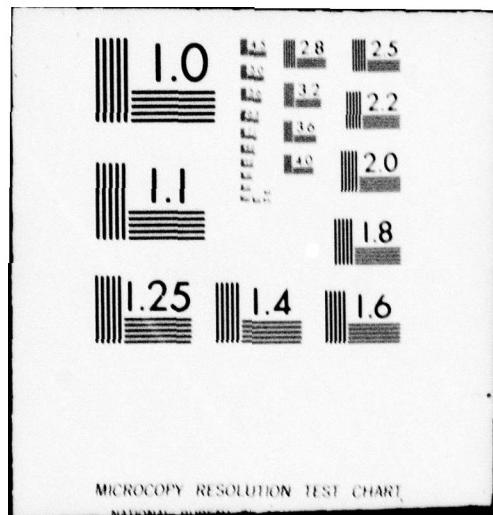
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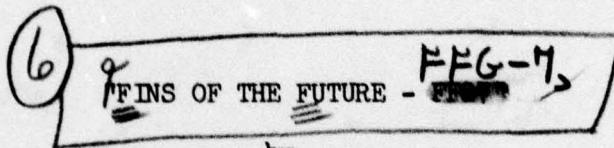
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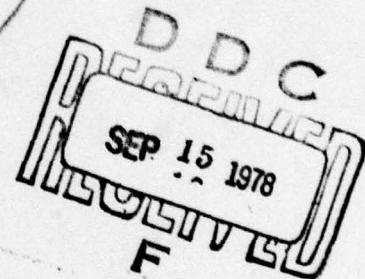
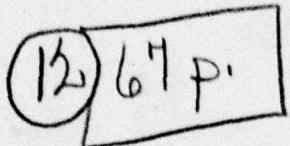
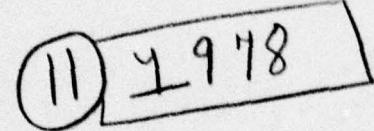
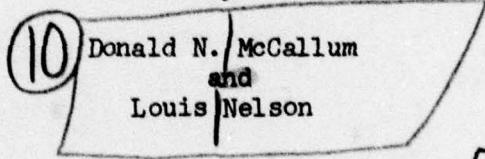
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ABSTRACT

This paper addresses the justification, design philosophy, system description and technical evaluation of the FFG 7 fin stabilization system - the fins of the future. These fins are to be installed on FFG 7 Class ships to provide increased mission effectiveness. Details of "lessons learned" are addressed and these are shown to be incorporated into this new design. Reliability and maintainability of the total system is stressed. Appendices provide useful design and specification tools.

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I. INTRODUCTION:

The advantages of roll stabilization by means of active fins have long been acknowledged by ship designers and operators. Since the patent for stabilizing fins was granted to John I. Thornycroft in 1889, followed by Dr. Motora's first independent installation on a Japanese ship about 25 years later, fin stabilizers have been used on passenger ships, ferryboats, naval vessels and a multitude of small pleasure boats. The famous Queens each employed two pair of large, retractable fins to ensure maximum passenger comfort. For the past twenty years, the U.S. Navy has employed fin stabilizers, mostly on combatant ships, generally with a high degree of success. Kehoe (1) conducted a Fleet survey in 1972, which emphasized the Fleet operator's enthusiasm for fin stabilizers. A recent ASE paper by Gatzoulis and Keane (2) documented afresh the rationale for fin stabilizers, while touching upon some of the reliability and maintainability problems. This present paper is, in some respects, a sequel to the above paper, insofar as it extends the paper into the step-by-step design of a specific fin system, custom designed for FFG 7 Class.

The objectives of this paper are three-fold:

- o to educate the engineering community on the substantive benefits of fin stabilizers;
- o to describe the rationale for installing fins on the FFG 7 Class;
- o to present a clear understanding of the technical design of the FFG 7 fin stabilization system.

(1) Indicates Reference Numbers

II. JUSTIFICATION FOR FINS

When the FFG 7 contract design phase was completed, a space and weight reservation was made for a single pair of fin stabilizers. At that stage it was not known if a fin system would actually be installed, only that fins were desirable and that space and weight allocations would be prudent.

In the fall of 1975, during naval exercises with the British and Canadians in the North Atlantic, weather conditions became severe. Waves of 20 to 25 feet were encountered frequently, accompanied by winds gusting up to 30 knots. The Captain of the USS PAUL (FF 1080) noted that his ship was unable to conduct critical operations and expressed his frustrations at the ship's motions and lack of helicopter handling equipment. Following this incident, a study was made to reassess the FFG 7's performance with and without FINS and with combinations of FINS and *RAST (Recovery Securing and Transversing System). The results of this study are shown in Table 1.

*Similar to Canadian BEARTRAP System.

TABLE 1
PREDICTED FIN AND RAST EFFECTIVENESS FOR FFG 7

HELO OPERATIONAL EFFECTIVENESS	IN (1) SEA STATE 5	IN (1) SEA STATE 6	IN (1) SEA STATE 7	ALL YEAR AVERAGE IN NORTH ATLANTIC
WITH BILGE KEELS (B.K.) ONLY	70%	20%	0%	50%
WITH B.K. AND FINS	100%	100%	30%	90%
WITH B.K. AND RAST (2)	100%	100%	50%	95%
WITH B.K., RAST AND FINS (2)	100%	100%	95%	100%

- NOTES:
1. Sea States 5, 6 and 7 are represented by significant (average of the one-third highest) wave heights of 10.2 feet, 16.9 feet, and 30.6 feet, respectively.
 2. Assumed that Helo Ops with RAST cannot be conducted at significant roll angles more than $\pm 13^\circ$.

The technique used to obtain Table 1 predictions is explained in detail in Appendix 1. It is based upon Helo Operational Effectiveness Limitations for rolling, as described in reference (2) and reiterated herein:

Roll motions in excess of 5 degrees significant single amplitude will severely limit loading of aircraft ordnance and aircraft handling.

Principal Ship Characteristics are given in Table 2 below:

TABLE 2
SHIP PARTICULARS

Displacement,	3663 tonne (3605 long tons)
Length Between Perpendiculars, L_{pp}	124.4 metres (408 feet)
Beam, B	13.7 metres (45 feet)
Draft, T	4.52 metres (14.84 feet)
Metacentric Height, GM^*	1.00 metre (3.28 feet)
Natural Roll Period, T	9.40 seconds
Center of Buoyancy Below Metacenter, \bar{BM}	3.86 metres (12.65 feet)
Block Coefficient, C_B	0.463
Midship Area Coefficient, C_X	0.75
Longitudinal Center of Gravity, LCG	0.009 L_{pp} (Aft of Midship)
Vertical Center of Gravity, KG*	5.82 metres (19.09 feet)

*Corrected for free surface.

Based upon considerations of Table 1, and also upon considerations of the many other benefits (2) of FINS, such as

- o Improved UNREP Capability
- o Improved Weapon/Sensor Performance
- o Reduced Wear on Equipment
- o Increased Speed in Waves
- o Improved Crew Performance
- and o Increased Crew Safety

the decision was made in the fall of 1976 to install FINS on FFG 7 Class ships, beginning with the FY 79 acquisitions.

III. DESIGN PROCEDURE FOR FINS

1. DTNSRDC Studies - In July 1976, DTNSRDC performed an investigation for FFG 7 fin size and performance (3). The conclusions of this study were as follows:

- o One pair of active, non-retractable fin stabilizers, 55 ft² area each side, should be considered as the means of stabilizing the FFG 7;
- o There should be no bilge keels aft of the fin stabilizers, but that the bilge keels should be extended forward to about station 6.7 (1/3 of ship length);
- o With the fins and bilge keels as outlined above, the NAVSEC helicopter operational criterion would be satisfied up to sea state 6; in addition, the fins are predicted to meet the U.K. Navy criterion.

Figures 1 and 2 show salient results of reference (3) studies.

2. Further Consideration - However, NAVSEC decided to increase fin size from 55 ft² per fin to 60 ft² for the following reasons:

- o Roll Reduction at Slow Speeds - To operate helicopters in high sea states 5, 6 and 7, due to the prevailing wind, the ship often has to slow down in order to limit the wind over the deck. Thus, the ship has often to operate at speeds of 8 to 12 knots. By extending the maximum fin angle from 24° to 28° and by increasing the fin area to 60 ft², more effective roll reduction would be achieved in these high sea states, combined with slow speeds.
- o Possible Increase in Ship's Displacement - The FFG 7 displacement is currently around 3,700 long tons. However, there are indications that this weight may increase due to building margins being exceeded. Also, it is well known that naval vessels do increase in displacement as a function of time. A conservative ship weight of 4,000 tons was used for maximum fin design conditions.

3. Fin Wave Slope Capacity - The "wave slope capacity" of a fin stabilizer system is, by definition, the angle of heel which the ship would theoretically assume, when acted upon by the fin stabilizers at their maximum angle of attack, at a specified ship speed.

Assuming the lift per fin is L
then heeling moment due to fins is 2L.x
and since from equilibrium of forces

$$\overline{GM} \sin \theta = 2L.x$$

$$\text{then } \theta = \text{finwave slope capacity arc sin } \frac{2L.x}{\overline{GM}}$$

(where x = arm from center of pressure of fin to CG,
and assuming one pair of fins)

A general rule of thumb for commercial and Navy practice for wave slope capacity is 5° at cruising speed of the ship; and since the lift

$$L = \frac{1}{2} V^2 S C_L \quad \text{where } S = \text{fin area, one side}$$

$V = \text{ship speed, f.p.s.}$
 $C_L = \text{lift coefficient}$

C_L is obtained from reference (4) for a fin of characteristics shown in Table 4, $\frac{dC_L}{dB} = 0.043$

But the "effective wave slope capacity" reflects the actual stabilization moment obtained from the fins, considering the degradation effects of boundary layer and bilge keels (20% and 8% respectively).

$$\text{Thus effective } \frac{dC_L}{dB} = 0.043 \times (1.0 - 0.28) = 0.031$$

This figure checks well with previous calculations for full-scale fin lift on FF 1052 Class ships.

A comparison of effective wave slope capacity is shown in Table 3, for a ship speed of 20 knots, and indicated that the FFG 7 fins compare well with recent British and U.S. fin designs.

TABLE 3
COMPARISON OF EFFECTIVE FIN CAPACITY

SHIP	DISPLACE-MENT (LONG TONS)	FIN AREA EACH SIDE (SQ. FT.)	EFFECTIVE CAPACITY (DEGREES)
FF 1052	4160	75	3.06°
FFG 7	4000	60	3.65°
HMS CHARYBDIS	2854	40	3.65°
TYPE 42 FRIGATE	3500	About 71	3.0°

For the British ships shown in Table 3, a $\frac{dC_L}{dB}$ value of 0.025 was used, since their fin aspect ratio was about 0.5 (5).

4. Fin Planform - The details of the FFG 7 planforms are as follows, in Table 4, and are illustrated in Figure 3.

TABLE 4

AREA = 60 ft ²
ASPECT RATIO = 1.0
TAPER RATIO = 0.45
SWEET ANGLE = 11 Deg
SECTION: NACA 0015 FAIRED TIP

The fin section was selected as NACA 0015 based upon good cavitation performance of this section and the fact that it has been tested extensively (4). The tip section is faired to minimize tip vortices. The leading edge is swept back moderately in order to shed debris, while the high taper ratio (of 0.45) puts the center of pressure close to the hull, thereby, reducing fin stock bending moments. In addition, the high taper ratio helps to unload the tip, thereby, reducing tip vortices. The planform thus described is a result of past experience and testing and is considered a good compromise between low noise characteristics and good lift performance.

5. Maximum Fin Operating Angles - Figure 4 shows the maximum design operating angles of the fins as a function of ship speed. Two modes of operation shown in Figure 4 are:

- o Normal operating mode
- o Minimum cavitation mode

The normal operation mode would be used under normal conditions, where maximum crew comfort and ship mission effectiveness is desired. Where the ship would be in a combat environment, the fin control would be switched to "minimum cavitation" mode, and reduced fin performance would be accepted. However, for the FFG-7 fin design, even in the "minimum cavitation" mode, the NAVSEC helicopter criterion will be met up through sea state 6 beam seas. (See Figure 1)

6. Hydraulic Power Considerations - Now we have settled upon a fin size, configuration and range of angular travel as it varies with ship speed. The next consideration is the size and speed of the hydraulics

which move the fins from side to side. Based upon good empirical practice (6), the time from hardover on one side to hardover the opposite side, should be not greater than T_0 seconds. From Table 2, $T_0 = 9.4$ seconds.

For FFG 7 calculations, 9.0⁶ seconds was selected as a slightly more conservative number.

Hence fin excursion time = 1.5 seconds

Appendix 2 outlines the method used by Scherer (7) to predict the torque generated by active fins in a ship rolling sinusoidally. Using a U.S. Navy computer program based upon this method, predictions of torque, fin loading and tiller horsepower were obtained. These are shown in Figure 5, and represent the maximum operational values for design purposes.

7. Maximum Design Conditions - A fin stabilizing system must be designed to withstand unusual a "worst case" situations. The FFG 7 fin system has been designed for a "casualty ahead" condition, where the fins would be pushed to their maximum angle to attack by the hydraulic actuators, with the ship at maximum speed.

IV. SYSTEM DESCRIPTION

The total fin stabilizer system is composed of three basic elements; the fin subsystem, the hydraulic subsystem, and the control subsystem.

Figure 9 shows how these three subsystems interface with one another. A brief overview is given below; for the reader requiring more detailed information, Appendix 3 is included.

1. Fin Subsystem - There will be two fin subsystems per ship, one port and one starboard. These are composed of fins, stocks, bearings, seals and related items. For the FFG 7 design, it was required that each fin subsystem should be a packaged, modular unit, mounted on a single foundation. Figure 7 highlights the most important parts of the fin subsystem, these are:

1. Fins
2. Fin stocks
3. Fin stock bearing assemblies
4. Fin stock seals and packings
5. Air emission systems
6. Tiller assemblies
7. Grease supply systems
8. Support structure and mounting bedplates

The fin subsystem is designed so that it is able to be installed from outside the hull, with the ship drydocked. This facilitates installation procedures, particularly in a retrofit such as FFG 7. Plans are underway to prepare the foundations inside the ships, so that installation may be made by means of a single shell plating cutout.

2. Hydraulic Subsystem - There will be an independent hydraulic subsystem for each fin subsystem; one port and one starboard. Each subsystem is composed of hydraulic power units, actuators, tanks, and related items, for positioning the fins in accordance with command signals from the control subsystem.

The hydraulic subsystem is shown diagrammatically in Figures 7 and 8 and its principal components are:

9. Main hydraulic pump(s)
10. Hydraulic actuators
11. Electric motor(s) and controller
12. Pump-motor mountings
13. Grease supply systems
14. Tanks

These components and others are specified in detail in Appendix 3.

3. Control Subsystem - There is only one control system per ship, which operates both port and starboard units in synchronism. It is composed of a digital processor, controllers, control panels signal transmission lines, and related items, to provide fin positioning command signals to the hydraulic subsystems.

The control subsystem is shown diagrammatically in Figure 10. Its principal components are:

1. Bridge control unit (BCU)
2. Central control unit (CCU)
3. Central processor unit (CPU)
4. Auxiliary sensor unit (ASU)
5. Local control unit (LCU), port
6. Local control unit (LCU), starboard
7. Angle transmitter unit (ATU), port
8. Angle transmitter unit (ATU), starboard

These components are specified in detail in Appendix 3, and details of BCU, CCU and LCU are shown in Figures 11 through 13.

V. LESSONS LEARNED - GOOD DESIGN THROUGH EXPERIENCE

The U.S. Navy currently has six different fin stabilization systems in a variety of frigates (primarily, the FF 1052 Class). All of these systems have exhibited reliability and maintainability problems caused by design deficiencies, component obsolescence, poor logistics support, and inadequate technical manuals. Many of the imperfections existing in the current equipment can be avoided in future systems by learning from past mistakes and by avoiding known design deficiencies. Experience has proven that the problem areas summarized below adversely affect the operability, maintainability and reliability of the fin stabilizer systems.

1. Mechanical and Hydraulic Problems

- a. Fin shaft anti-friction bearings have become corroded and unserviceable due to exposure to sea water. Anti-friction type bearings must be adequately protected from sea water exposure.
- b. Fin shafts in way of phenolic sliding surface type bearings have become severely corroded due to sea water. The fin shafts in this type of application should be sleeved with a corrosion resistant material.
- c. Hydraulic rams are easily scored and nicked due to the lack of a protective covering. This causes excessive ram packing leakage, hydraulic oil loss, and increased maintenance. Rams and cylinder piston rods should be protected from damage and dirt.
- d. Hydraulic system filters are often found to be collapsed and/or bypassing. The use of adequately sized, non-cleanable, national stock system supported elements, and suitable housing with integral bypass valves and differential pressure indicators is highly desirable. This will allow for proper protection of the system from wear and will greatly facilitate logistic support and maintenance.
- e. Control pump suction strainers should be readily accessible and should incorporate some means of monitoring their condition.
- f. Main system pumps have been severely damaged and have required complete overhaul due to inadequate replenishment pump flow. Means should be provided to protect the main pump from cavitation.
- g. Hydraulic pumps should be rated for continuous operation at maximum operating pressure. This will greatly enhance pump reliability and life expectancy.
- h. The lack of logistic support for obsolete components has greatly hampered the maintenance of the systems. The use of special, non-standard, long lead time equipment should be avoided.

i. Adequate pressure, vacuum and temperature gages and other instrumentation should be incorporated to facilitate rapid troubleshooting of any and all portions of the system.

j. Severe damage to ram assemblies and pumps has been experienced when rams overtravel. Adequate safeguards should be built into the equipment to protect it from damage when some part of the system malfunctions.

k. Copper crushable stops have caused system damage, both before and after being crushed, due to inadequate attachment. Copper crushable stops should be securely attached, easily replaceable by Ship's Force, and located away from sensitive components which might be damaged as a result of a loose or deformed copper stop.

l. All hull gland and ram packings should be split-type and should be replaceable with a minimum of system disassembly, and without the need to drydock the ship.

m. A means of obtaining hydraulic fluid samples that are truly representative of the active system fluid should be provided, e.g., sampling valves.

n. Hydraulic oil coolers except where the tank acts as a cooler, should be in accordance with MIL-C-15370 to minimize the possibility of contaminating the hydraulic system with sea water.

o. Hydraulic system filter maintenance should be possible without having to drain down the system and/or head (expansion) tank.

p. The overall system maintenance requirements should be minimized and as simple as possible.

2. Electrical and Control Problems

a. Inadequate logistic support of control system components has severely hindered the maintenance and repair of the systems. Parts availability should be given high priority, both as to stocking levels of repair parts and avoidance of using non-standard or special components that have limited availability.

b. The use of MS connectors vice hard wiring would greatly facilitate maintenance and minimize the chance of miswiring during maintenance and repair actions.

c. Troubleshooting, testing and maintenance of systems with control components located within the fin shaft bore has been difficult. This type of arrangement has caused significant system downtime. All components that may require maintenance should be readily accessible.

d. High humidity and moisture have caused numerous control system components to corrode and malfunction. Vulnerable components should be sealed or otherwise protected from the environment. Controls should also be located away from such devices as steam traps, fire pumps and water lines.

e. Bearings in control system components, e.g. rate gyros, have frozen due to lack of use. This type of malfunction can exist for a long time without Ship's Force awareness. System design and equipment operating instructions should provide for frequent and regular operation of all such bearings.

f. The lack of adequate training and technical documentation has adversely affected the operation and maintenance of the systems. New equipment should make the greatest possible use of system self-test, automatic trouble diagnosis and modular construction. Operator training and adequate documentation should be provided.

VI. RELIABILITY AND MAINTAINABILITY - THE NAME OF THE GAME

In view of the generally unsatisfactory RMA characteristics of the fin stabilizer systems procured by the U.S. Navy in recent years, RMA considerations will be of paramount importance during the design, fabrication, testing and evaluation phases of the fin stabilizer procurement program for FFG 7.

1. Reliability and Maintainability (R and M) Design Requirements - A comprehensive R and M program in accordance with MIL-STD-785 and MIL-STD-470 will be implemented and the R and M requirements will be imposed on both contractors and subcontractors. The quantitative R and M requirements are as follows:

- o Mean-Time-Between-Failure (MTBF) for the system shall be not less than 1,000 hours.
- o Mean-Time-Between-Failure (MTBF) for that portion of the system that is not repairable aboard ship (i.e., requires tools, test or support equipment not aboard FFG or requires tender or shipyard repairs) shall be not less than 6,000 hours.
- o Mean-time-To-Repair (MTTR) for the system shall not exceed 1.0 hours for all repair actions to be performed at the organizational level of repair (repairable by ship's crew underway)
- o Maximum Repair Time: 90% of the repair actions shall not require greater than 3.0 hours at the organizational level of repair.

2. R and M Failure Reporting Analysis and Corrective Action (FRACA) -

The contractor will establish and implement a FRACA program to determine failure patterns and trends, cause(s) of failure and to implement and verify necessary corrective actions. The FRACA program will pertain to subsystem level and system level factory tests and all testing at Government facilities. The FRACA program shall consist as a minimum of the following elements:

- o Documentation of all failures, malfunctions, design defects, deficiencies, discrepancies, technical problems and other corrective maintenance actions; failure analyses, corrective actions.
- o Analysis of each relevant failure malfunction, design defect, deficiency, discrepancy, technical problem or other corrective maintenance action to determine cause
- o Determination of problems, failure patterns and trends requiring corrective action
- o Determination of required corrective action and verification of the adequacy of the corrective action implemented.

3. Maintenance Engineering Analysis (MEA) - The contractor will conduct a MEA of the system in accordance with MIL-M-24365 to determine the most effective and efficient procedures for performing maintenance. The MEA report will include the effects of failures on safety and the precautions to be taken during maintenance to eliminate or reduce safety risks.

4. Corrective Maintenance - Corrective maintenance downtime will be reduced by designing for rapid and positive detection of malfunction of fault corrections, and rapid positive verification of corrective action taken. The control subsystem shall contain automatic fault isolation. For the Central Processor Unit (digital electronics) fault isolation will be to within five modules 95% of the time and within 12 modules 100% of the time. For each Local Control Unit (analog electronics), a digital voltmeter and selector switch arrangement will be provided to monitor the output voltage of operational amplifiers and all other test locations considered to be significant for troubleshooting purposes. A schematic diagram with all test locations clearly identified by "OUTPUT SELECTOR" switch positions AA, AB, ...KL, LL will be permanently attached to the inside cover of each Local Control Unit.

5. Special Maintenance Features - Special maintenance features peculiar to the fin stabilizer system are as follows:

a. Fin Assemblies - Collar keys for fin carriers, glands, retainer rings and packings will be split as needed to permit maintenance without unshipping the fins.

b. Fin Stock Bearings - Means will be provided for the installation, assembly and removal of all fin stock bearings. The inboard and outboard bearings will be grease lubricated, and accessible fittings and vents will be provided for pressure grease lubrication.

c. Fin Stock Seals - The inboard fin stock packings will be renewable and adjustable as necessary, from within the ship, without drydocking the ship, and without flooding of the compartment. All glands for compression type packings will be readily accessible from inside the ship to take up on follower units. The packing in the inboard fin stock seal stuffing box will be renewable without removing the fin and fin stock, or the fin stock tiller. The design will be such that each of the above maintenance actions can be accomplished by Ship's Force.

d. Motors and Pumps - The arrangement of the hydraulic subsystem will permit removing the motors and pumps with a minimum delay and without disturbing other components. It will be possible to replace pump drive shaft seals without removing the pump or motor.

e. Lifting Fixtures - A recessed lifting fittings or hole will be provided in at least two corners of each fin for handling purposes. The fitting or hole will be made flush with the finshape by means of a suitable filler or screwed insert to minimize cavitation. Appropriate identification of these lift points will be marked on the fin and shown on the appropriate drawing. A lifting eye will be provided near the top of the fin stock to facilitate shipping and unshipping the fin and fin stock when the ship is in drydock. These lifting fittings will be designed to safely support the entire fin with the fin stock attached including the weight of water if the fin is completely flooded.

A minimum of three equally spaced tapped holes for lifting eye-bolts shall be provided in both faces of the roller bearing outer race for the inboard bearing, and in both halves of the stave bearing bushing flanges of the outboard bearing, for bearing installation and removal. The outboard bearing assembly will be installed on stepped bushings, suitable for unshipping, for bearing restaving and boring.

6. Operating Life - The system will have a minimum design life of 20 years with a 30% duty cycle, and will be designed for 6,000 hours of operation or 6 years between overhauls, whichever is greater. The main hydraulic pumps will have a minimum design life of 6,000 hours of operation. Predictions for the service life of the pumps will be based upon pump contaminant sensitivity tests.

7. Personnel - Required manning will be composed of presently available Navy skills as identified in NAVPERS 18068. The mechanical portion of the system will be designed for maintenance by no more than two enginemen. The electrical portion of the system will be designed for maintenance by no more than two electrician's mates.

VII. QUALITY ASSURANCE

1. Quality Control System - The quality program will be in accordance with MIL-I-45208.

2. Contractor Factory Tests - The contractor will perform the following tests prior to submitting a complete fin stabilizer system to the Government:

a. Subsystem qualification examinations and tests. Subsystem qualification examinations and tests will be performed on each subsystem to be delivered by the contractor. These will verify that each subsystem is in accordance with those requirements which can be tested apart from the other subsystems.

b. System qualification examinations and tests. System qualification examinations and tests will be performed to verify the integral system performance requirements.

Subsystem and system qualification examinations and tests may be conducted in parallel. All qualification examinations and tests will be performed in accordance with the examination and test plan.

3. Examination and Test Plan - The contractor will submit a written examination and test plan which contains the procedures, data forms, a list of test equipment to be used and the sequence of tests for performing inspection to the Government prior to beginning inspection.

4. Fin Subsystem Tests - The following examinations and tests will be performed on the fin subsystem.

- a. Surface Examination
- b. Shock

c. Air Tightness (fin only) - Following final welding and surface preparation, and prior to painting, each fin will be tested for tightness with air to a pressure of 10 psig. These tests will be conducted prior to filling with preservative coating.

d. Fairness - a check for fairness and smoothness will be made on each fin by use of a flexible wooden batten 24 inches long and 3/16 inch square cross section. Checks for deviations from the nominal contour will be made by means of chordwise half-templates extending 12 inches aft of the leading edge at a maximum of 12 inch intervals. Batten checks, will be made spanwise along the leading edge.

5. Hydraulic System Tests - The following examinations and tests will be performed on the hydraulic subsystem.

- a. Surface Examination
- b. Low Temperature Test
- c. High Temperature Test
- d. Vibration Test
- e. Inclination Test
- f. Hydrostatic Pressure Tests
- g. Airborne and Structureborne Noise Tests
- h. Shock Test
- i. Pump Contamination Tests

Pump Contamination Tests - The hydraulic pump contaminant sensitivity test will be performed on both the main and charge hydraulic pumps in accordance with the National Fluid Power Association (NFPA) recommended Standard T 3.9.18-1976. The test report will include pump contaminant tolerance profiles for 1000, 5000, and 10,000 hours based upon life defined alternatively as 5, 10, and 20 percent flow degradation.

6. Control Subsystem Tests - The control subsystem, will be subjected to examinations and tests to demonstrate compliance with applicable MIL-E-16400 specification requirements. The performance test will include all control subsystem performance tests that can be accomplished prior to system integration, and will demonstrate that the performance of each control, indicator and display function is within the functional specification requirements.

7. System Level Testing - The contractor will log a minimum of 50 hours of operation and/or tests at the system level prior to delivery to DTNSRDC. The following tests will be performed.

- a. Operation Check
- b. Steady State Voltage and Frequency
- c. Transient Voltage
- d. Transient Frequency
- e. Power Interruption
- f. Examination

Each unit comprising the fin stabilizer system will be examined to insure compliance with the applicable specifications with respect to surface examination, weight, size and marking.

8. Laboratory Testing - Upon completion of 200 hours of factory testing, which includes 50 hours at the system level, each of the two contractors will deliver their FSS to DTNSRDC Annapolis, where the Navy will test both units to determine their merits. Each FSS will undergo a rigorous reliability and maintainability (R&M) evaluation test. The reliability test will consist of 1000 hours of equipment operation in operating modes which are anticipated during in-service use. The Maintainability tests will consist of the performance of both corrective and preventive maintenance tasks to evaluate MTTR's and maximum repair times.

9. At-Sea Testing - The final testing effort will be conducted at-sea on the first available FFG after installation during PSA (i.e., for planning purposes FFG 8). The contractor who is awarded a contract to provide the production fin stabilizers will be required to implement all corrective action identified during the DTNSRDC testing effort on his FSS, and groom the equipment for installation on the FFG 8 for at-sea trials. The contractor will be authorized to procure selected long lead items prior to completion of at-sea trials, however, release for manufacture will be deferred pending successful completion of trials.

10. Production Tests - Production FSS will be subjected to the following examinations/inspections/tests:

a. Factory Acceptance Tests (FAT) - Each production FSS will be subjected to the following FAT's prior to delivery:

(1) Subsystem Level Tests

- (a) Fin Subsystem - Each fin stock/fin air circuit will be subjected to the Air Tightness Test.
- (b) Hydraulic Subsystem - Each hydraulic subsystem and its components will be subjected to the Hydrostatic Pressure Tests and the Subsystem Operational Test.
- (c) Control Subsystem - Each control subsystem will be subjected to the Subsystem Operational Test followed by the Vibration Test and another Operational Test.

(2) System Level Tests

Each FSS shall be subjected to a System Operational Test followed by a Burn-in/Failure Free Operating Test, another System Operational Test and an examination.

- b. Reliability Demonstration Test - A complete fin stabilizer production system (to be selected by the Navy) will be subjected to Reliability "R" demonstration test consisting of 1000 hours of operation with no (zero) allowable failures. Testing will be in accordance with MIL-STD-781. During the test, the FSS equipment will be subjected to the maximum loads, modes of operation and operational profile anticipated during actual use. Input voltage cycling will be performed at 90%, 100% and 110% of nominal voltage for equal periods of time at least once per day. Environmental conditions (temperature, pressure and humidity) will be existing test facility ambient conditions. Only that preventive maintenance (PM) and PM

frequency proposed by the contractor and approved by the Navy prior to the test will be performed during testing. Upon completion of the test, hydraulic components will be disassembled and examined for excessive wear, galling and fretting or other deterioration.

- c. Maintainability Demonstration Test - A production fin stabilizer system (to be selected by the Navy) will be subjected to Maintainability "M" testing. The procedures of the technical manual will be used in the performance of all "M" testing. Corrective maintenance repair times to be demonstrated will include times for fault detection, localization, isolation, remove and replace (or repair in place) and performance verification.

VIII. SUMMARY AND FUTURE PLANS

To summarize the amount of thought, work and engineering which has gone into the concept and implementation of the FFG 7 fin design is difficult, since so many different facets are involved. However, as mentioned in the Introduction, the principal objectives of this paper are to educate on fin stabilizers, to describe the rationale behind FFG 7 fins, and to present a clear understanding of the technical design of FFG 7 fins.

The rationale behind FFG 7 fin design and procurement can be summarized as follows:

- o Fly-off testing between two vendors, to select fin system.
- o Design fin system to incorporate all previous lessons learned.
- o Reliability and Maintainability - very strong emphasis.
- o Quality Assurance - testing of fins and subsystems both at Manufacturers' Plant, DTNSRDC/Annapolis, and at-sea.

Thus the FFG 7 fin system, comprising one pair of 60 ft^2 non-retractable, non-articulated trapezoidal fins, has been engineered to be a highly reliable system, incorporating all the latest technical advances, such as Standard Electronic Modules.

Future plans for the FFG 7 fin system are to perform at-sea testing on the prototype, prior to installation on the rest of the FFG 7 Class, commencing with FY 79 hulls. It is anticipated that the FFG 7 design will become a landmark for U.S. Navy fins of the future, and the specifications, drawings, and design philosophy will be extended into larger and smaller "standard" fin sizes. These sizes are anticipated to be 40 ft^2 and 90 ft^2 . This would cover ship displacements of about 2000 tons through 10,000 tons for a single pair of fins per ship.

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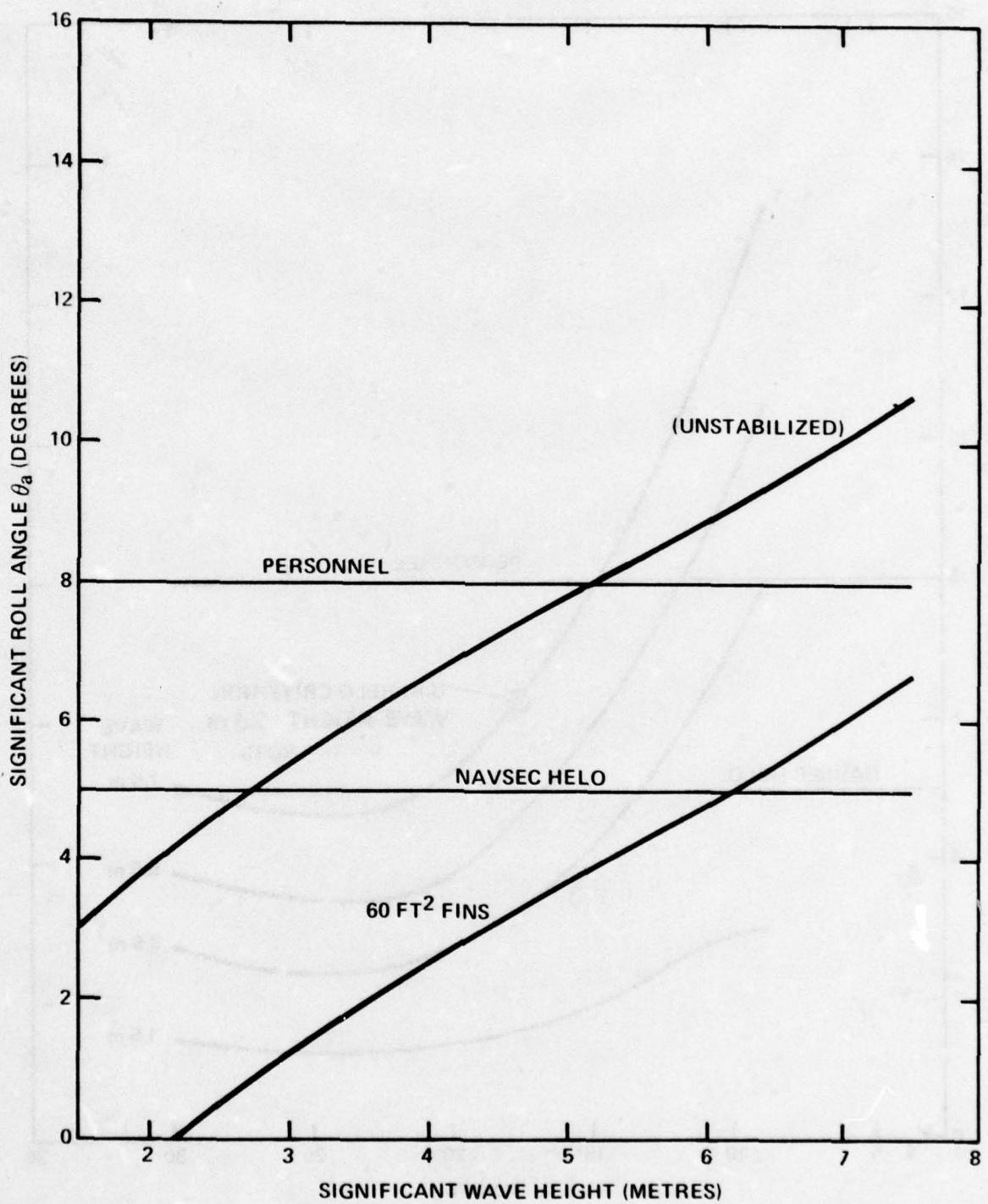


FIGURE 1 FFG-7 REDUCED NOISE OPERATION ADAPTIVE CONTROL 17 KNOTS

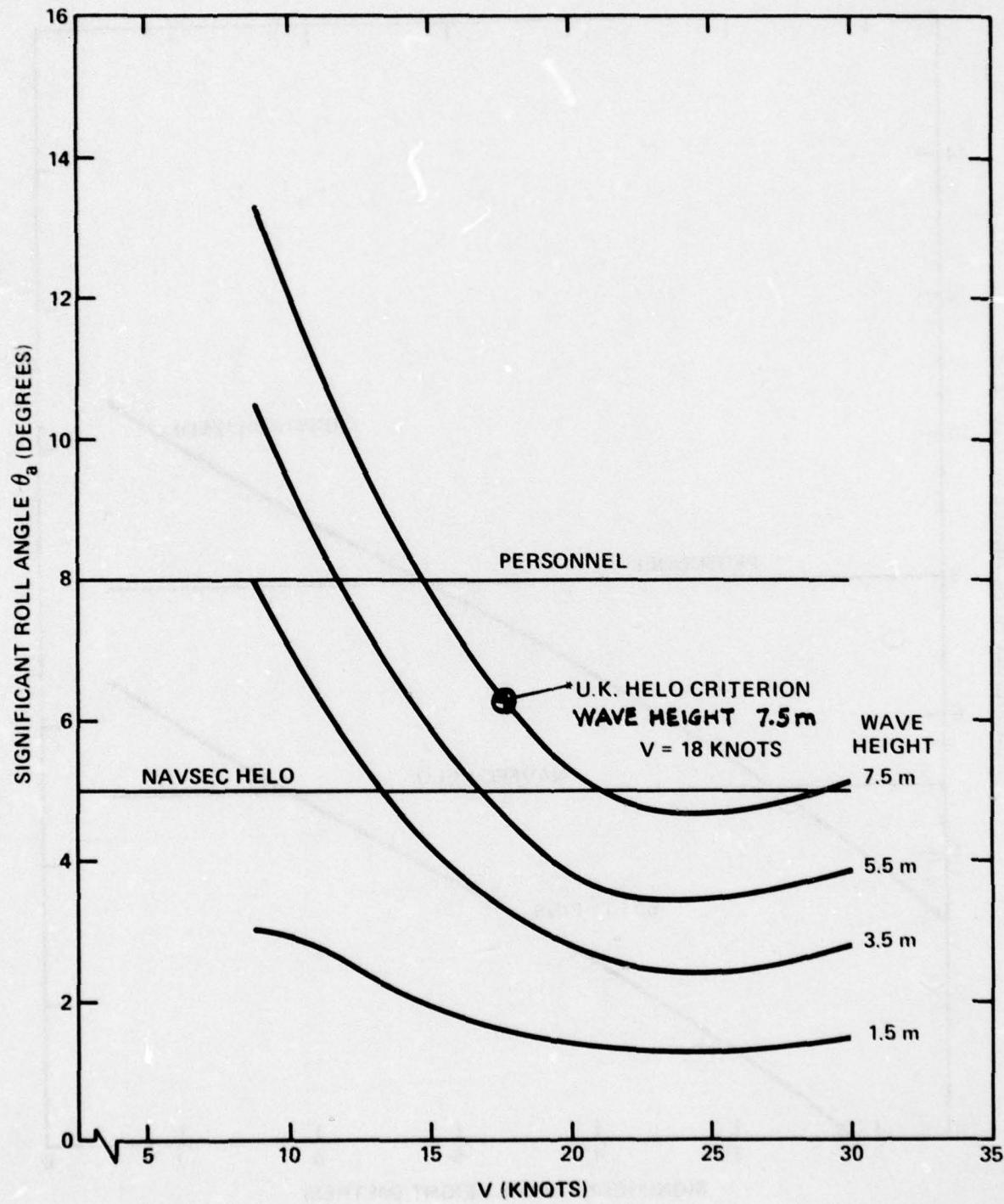


FIGURE 2 FFG-7 55 SQ FT FIN (ACTIVE) REDUCED NOISE OPERATION

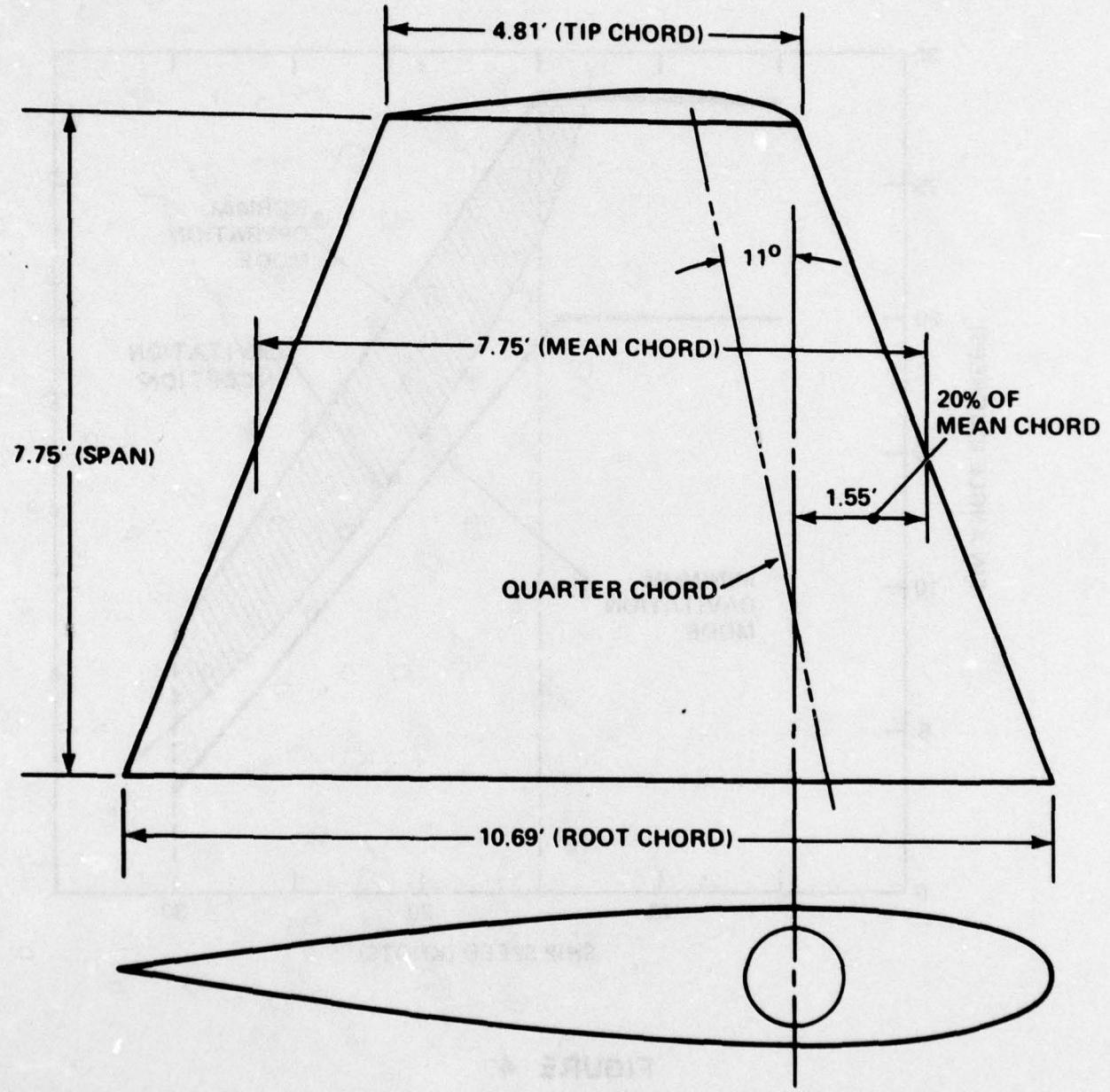


FIGURE 3 FIN PARTICULARS

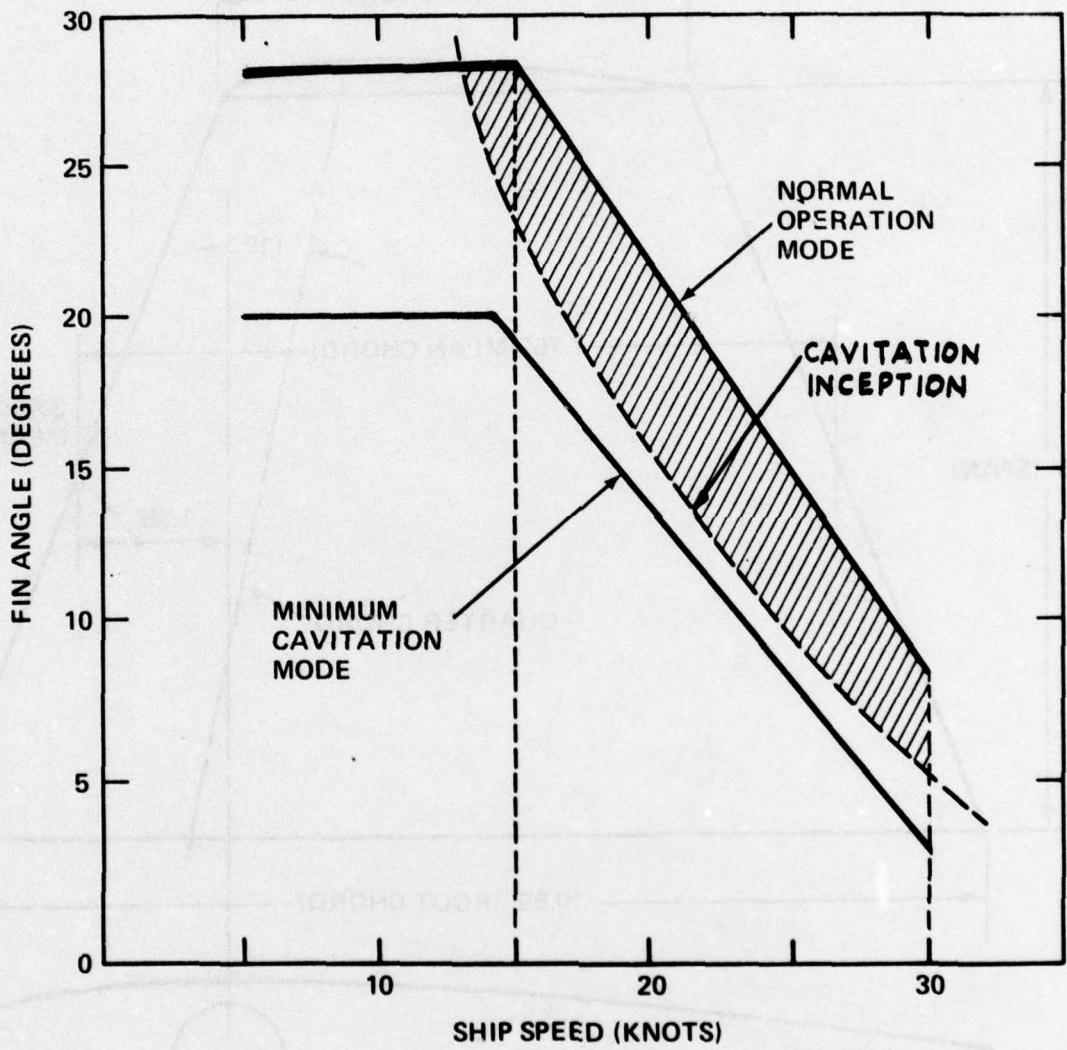


FIGURE 4
OPERATIONAL LIMIT ANGLES

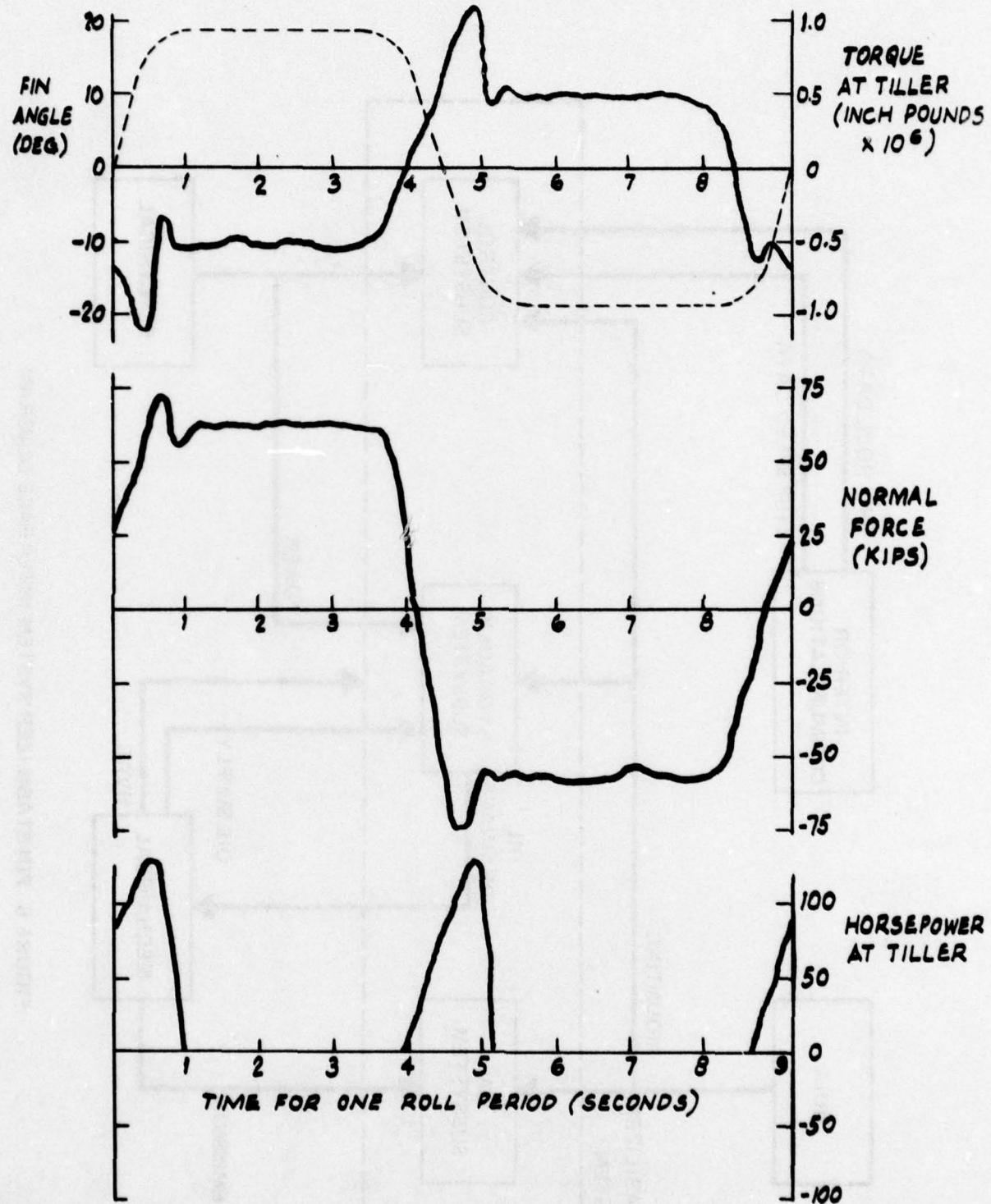


FIGURE 5
TORQUE, FORCE & HORSEPOWER

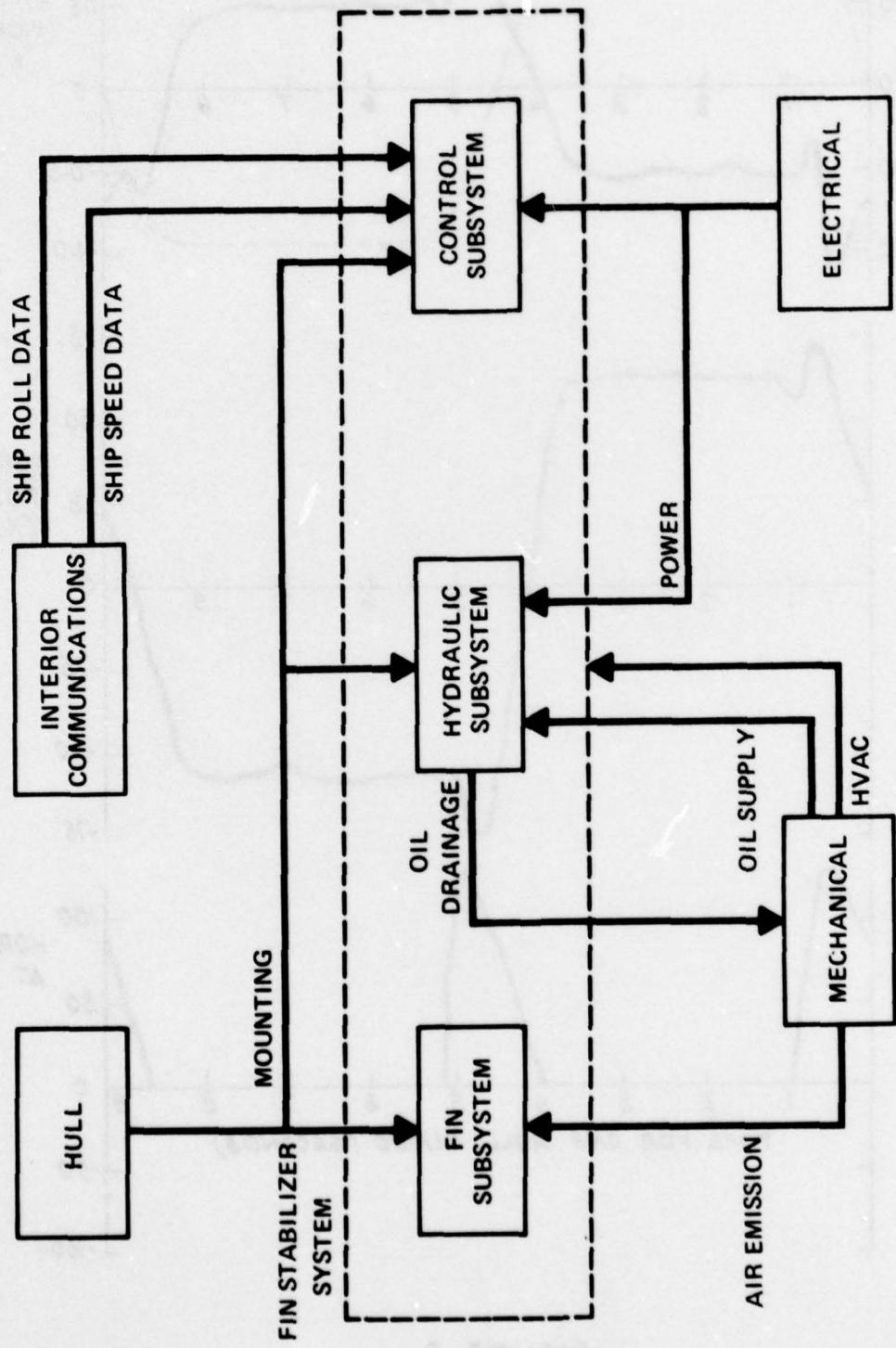


FIGURE 6 FIN STABILIZER SYSTEM INTERFACE DIAGRAM

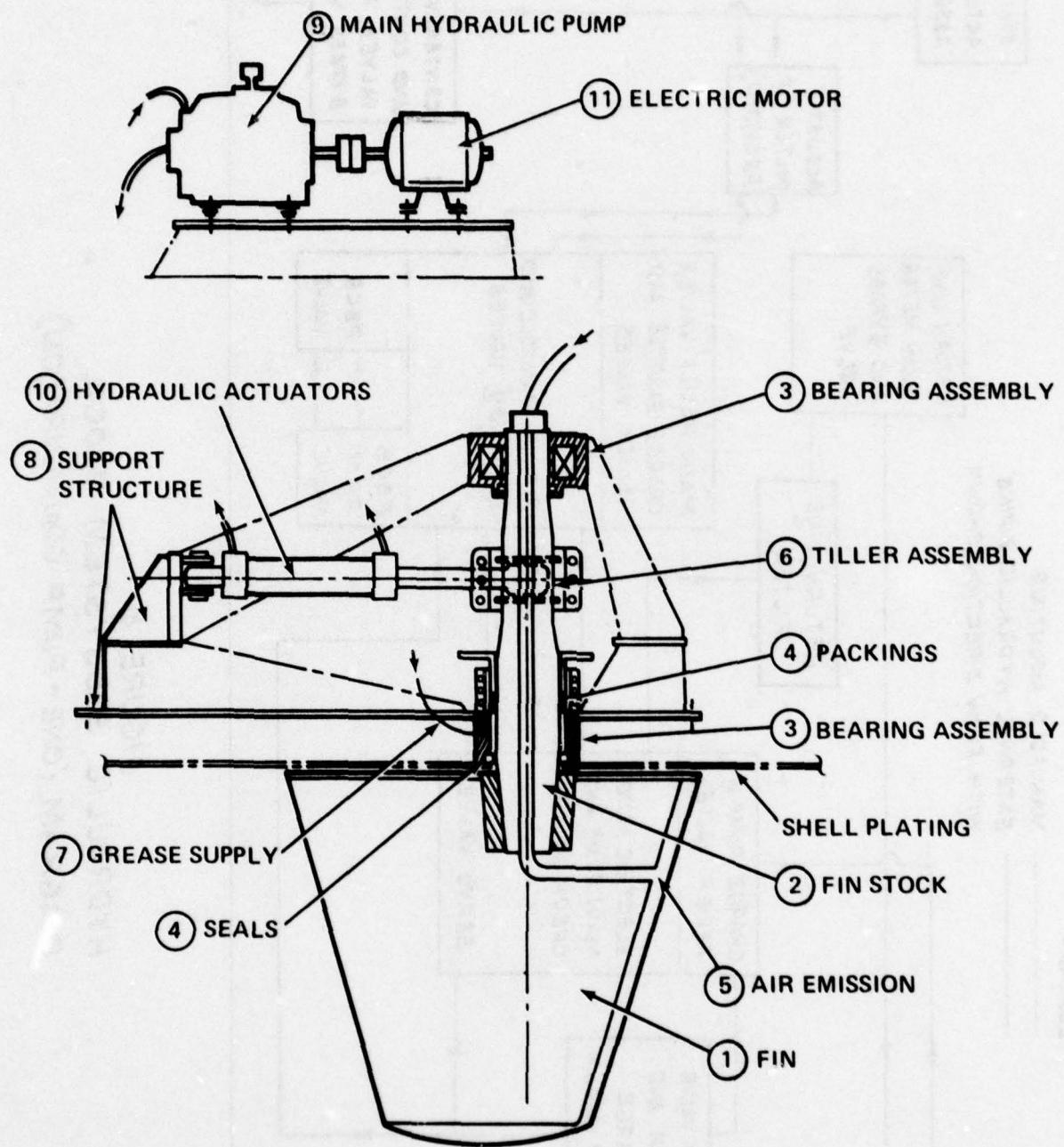


FIGURE 7 SCHEMATIC OF FIN AND HYDRAULIC SUBSYSTEMS

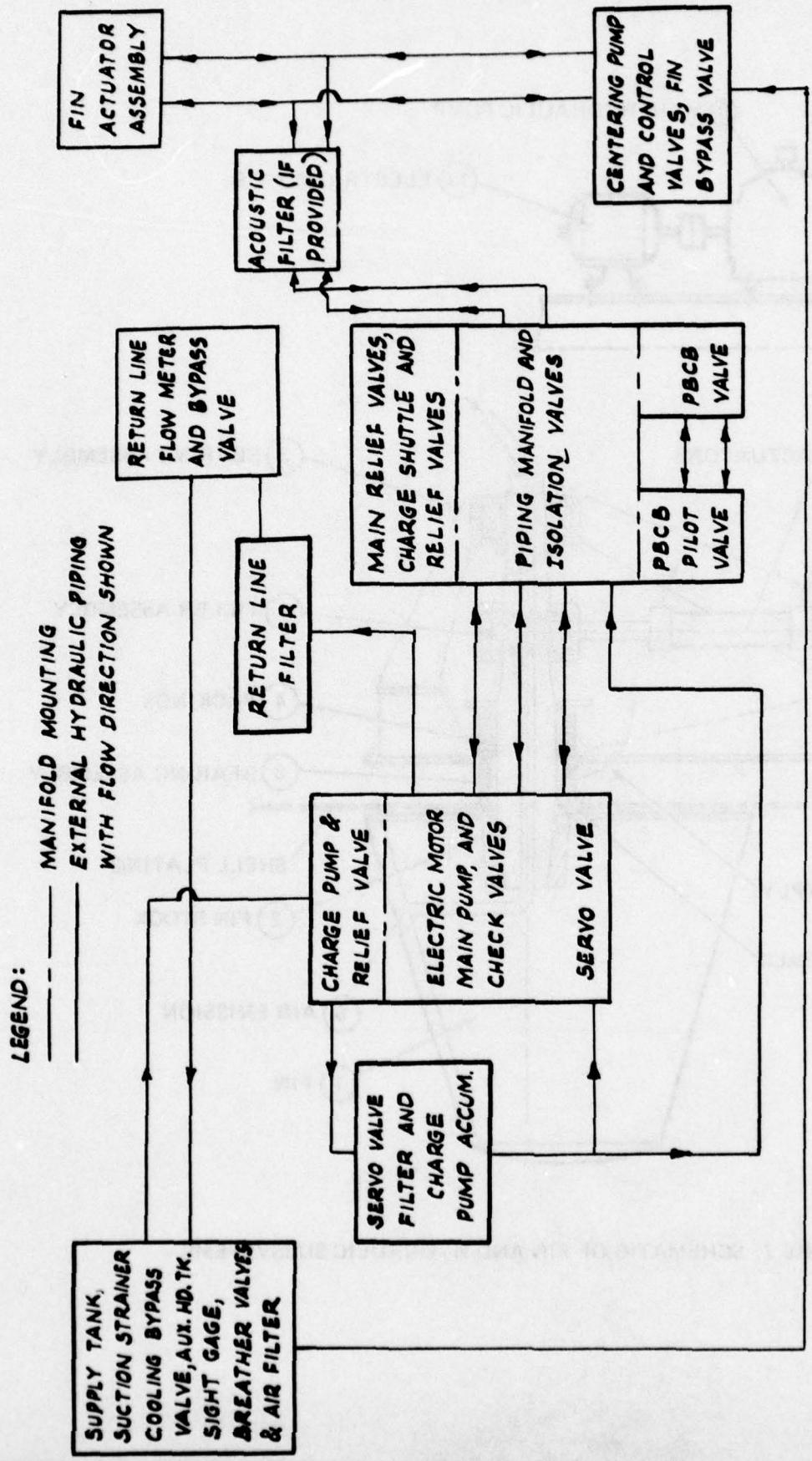


FIGURE 8
HYDRAULIC SUBSYSTEM BLOCK
DIAGRAM (ONE - PUMP CONFIGURATION)

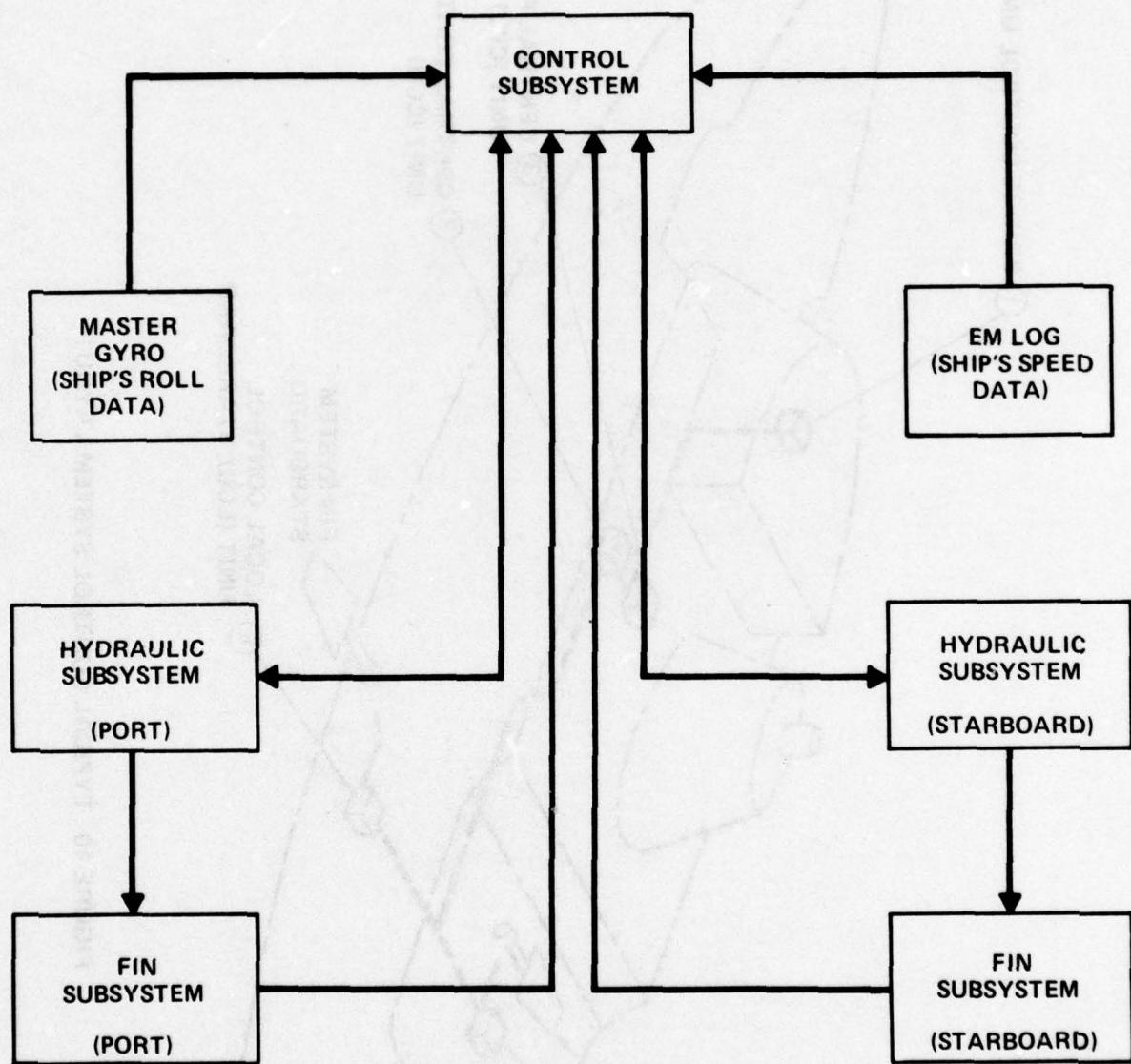


FIGURE 9 CONTROL SUBSYSTEM INTERFACES

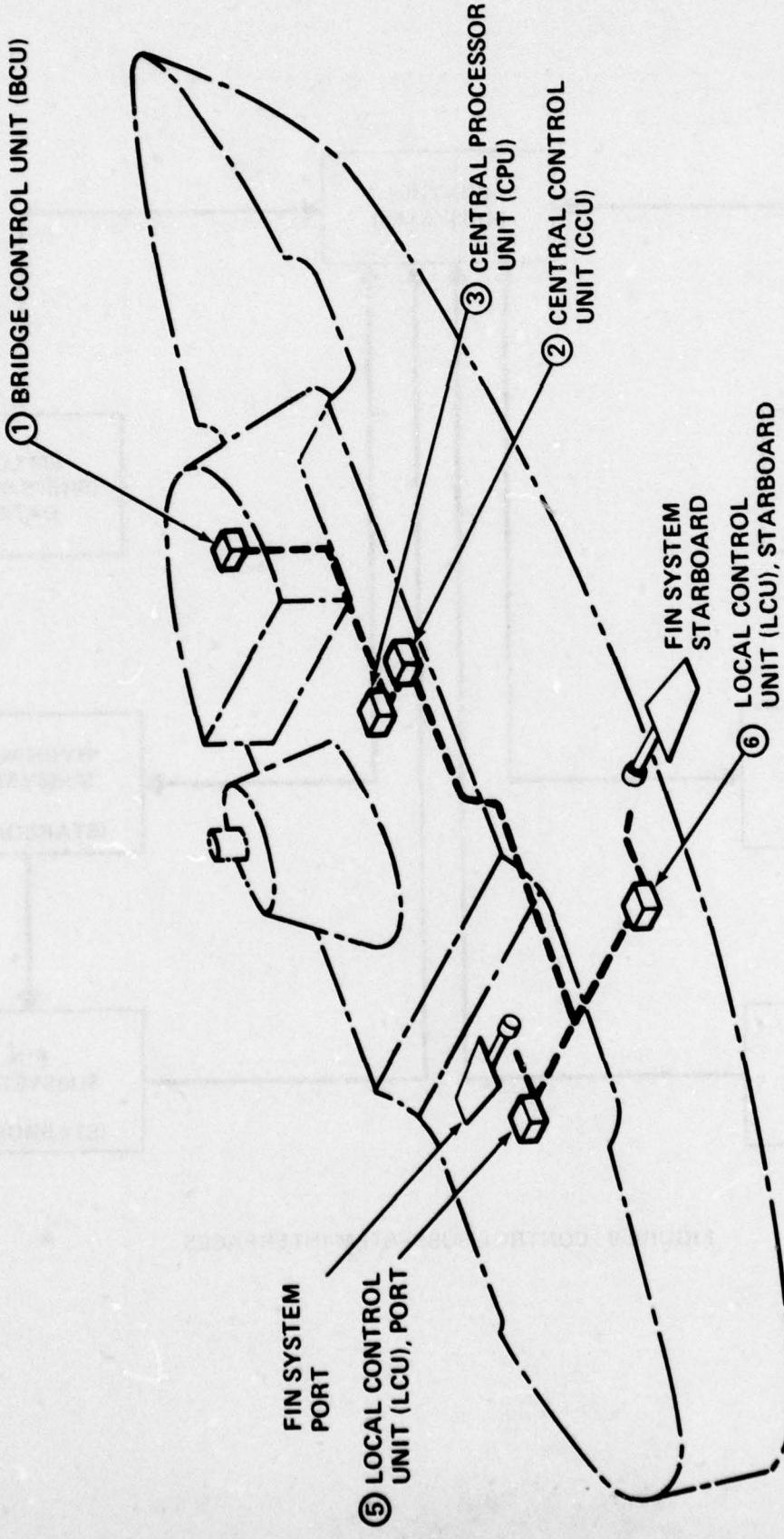


FIGURE 10 TYPICAL CONTROL SYSTEM LAYOUT

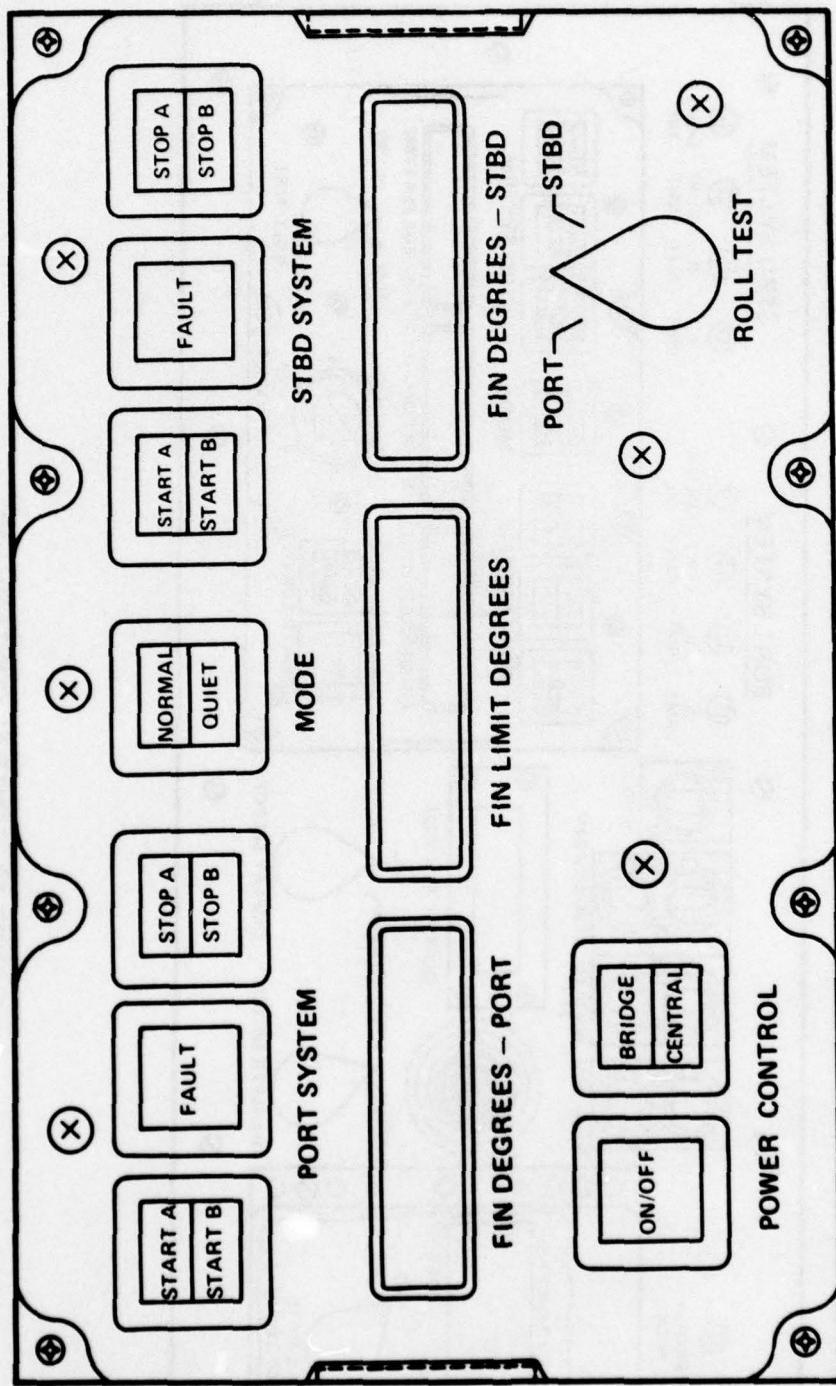


FIGURE 11 BRIDGE CONTROL UNIT

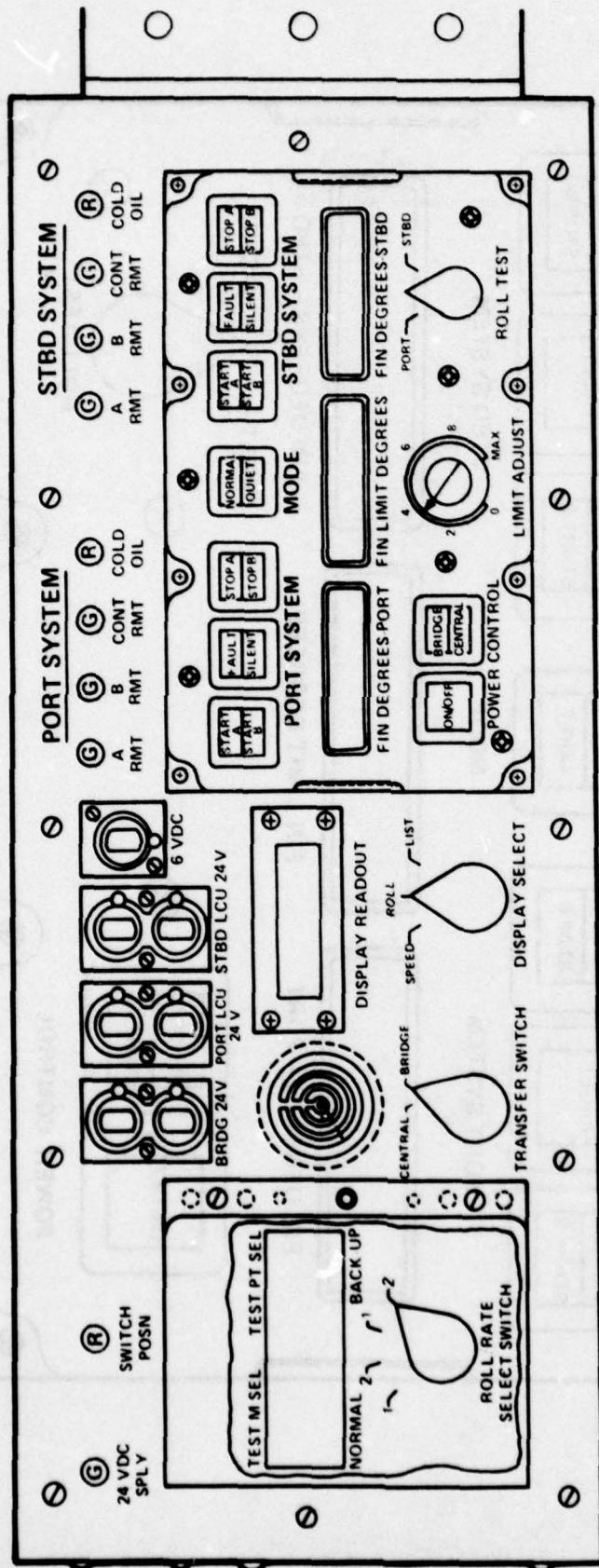


FIGURE 12 CENTRAL CONTROL UNIT

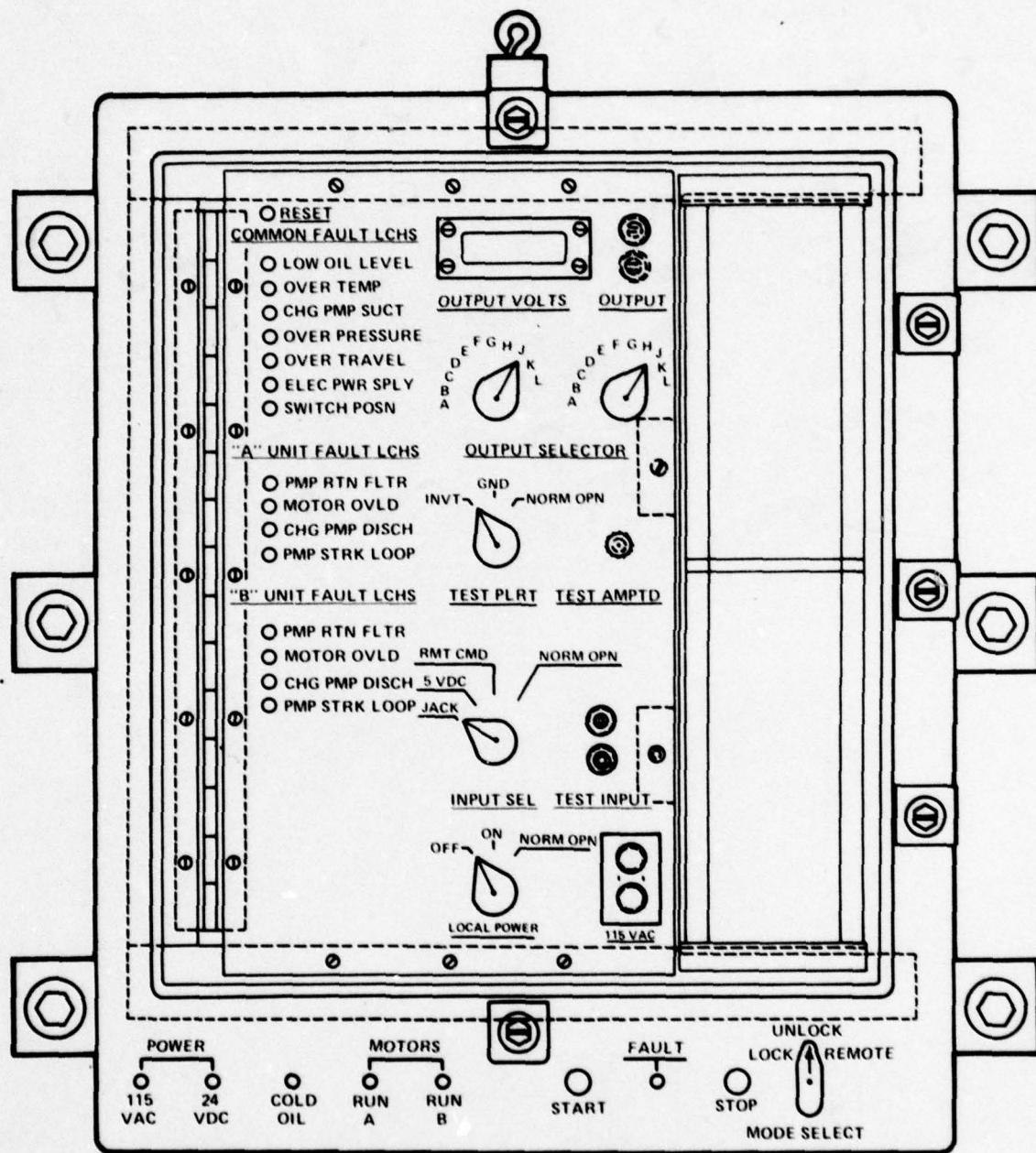


FIGURE 13 LOCAL CONTROL UNIT

APPENDIX 1 - TECHNIQUE FOR PREDICTION OF HELO OPERATIONAL EFFECTIVENESS

STEP 1

- o Prepare curves of roll angle vs. wave direction, for a range of sea states from 5 to 7, assuming short crested seas. Curves should be for ship with no fins (bilge keels only) and with fins.

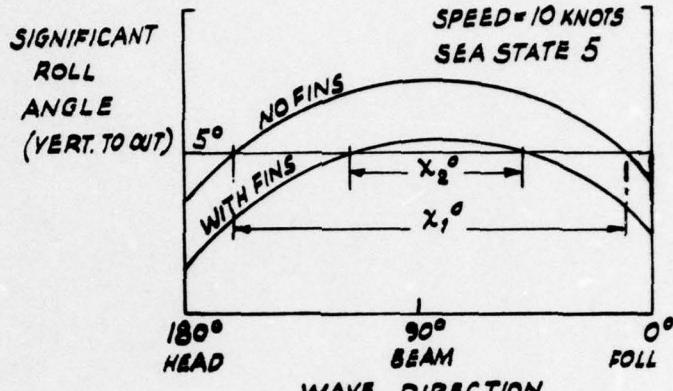


FIGURE A1

STEP 2

Using a suitable criterion (5° for helo handling), obtain the % of time when operations can be conducted.

$$\text{i.e. \% of time in sea state } 5 = \frac{180^{\circ} - x^{\circ}}{180^{\circ}}$$

at speed of 10 knots. 180° .

perform this calculation for the range of sea states, and ship speeds.

STEP 3

Draw curves of helo operational effectiveness vs. speed, for each sea state.

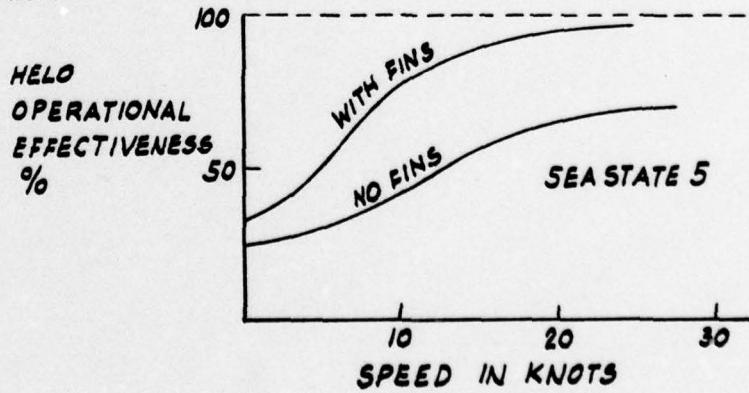


FIGURE A2

STEP 4

- o Obtain the ship's speed profile (example shown in Figure A3) and simplify into speed blocks, i.e.

<u>SPEED (KTS)</u>	<u>% OF TIME</u>
0 - 10	12
10 - 15	10
15 - 20	26
20 - 25.	41
<u>25 - 30</u>	<u>11</u>
ALL SPEEDS	100

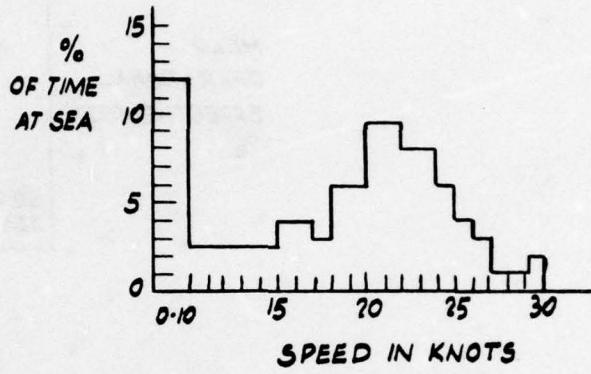


FIGURE A3

STEP 5

- o Obtain % of time that operations can be conducted in one sea state, considering speed profile of ship, i.e. for sea state 5.

- o Pick off mean values from Figure A2 at midpoint of speed blocks, i.e. for 15 - 20 knots, pick off value at 17.5 knots.

<u>SPEED (KTS)</u>	<u>% OF TIME</u>	OPERATIONAL EFFECTIVENESS		CUMULATIVE	
		NO FINS	WITH FINS	NO FINS	WITH FINS
0 - 10	12	30	45	4.5	5.4
10 - 15	10	50	80	5.0	8.0
15 - 20	26	60	90	15.6	23.4
20 - 25	41	65	95	26.0	39.0
25 - 30	11	70	100	7.7	11.0
TOTALS				58.8	86.8

These cumulative totals are used in assessing the fin performance at each particular sea state.

STEP 6

- o Draw curves of helo operational effectiveness vs. sea state.

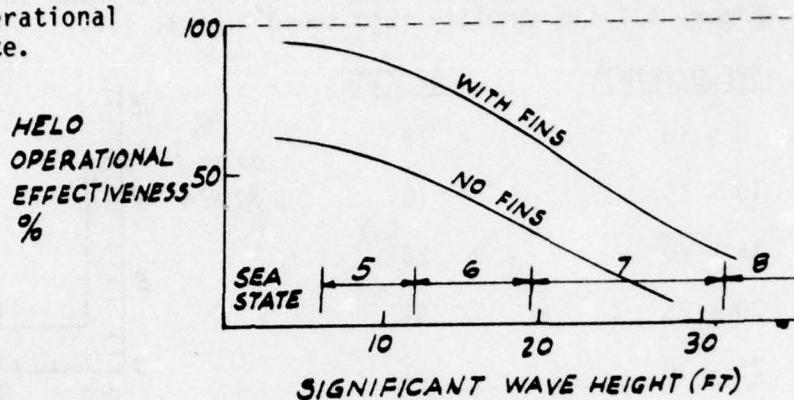


FIGURE A4

STEP 7

- o Introduce probability of occurrence of each sea state, in a selected area of the world, and arrive at a "all year average." NAVSEC practice has been to use the "all year North Atlantic" probability, since many ships conduct exercises in this area.

Thus we have:

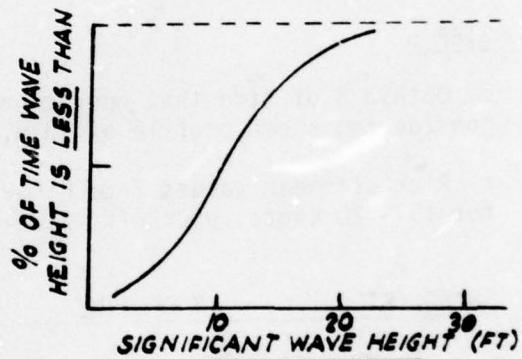


FIGURE A5

SIGNIF. WAVE HT (FT)	% PROB.,	OPERATIONAL EFFECTIVENESS		CUMULATIVE	
		No fins	fins	No fins	fins
0 - 10	50	0.60	0.95	0.30	0.48
10 - 15	33	0.50	0.85	0.16	0.28
15 - 20	11	0.35	0.65	0.04	0.07
20 - 25	4	0.20	0.55	0.01	0.02
25 - 30	2	0.10	0.35	.00	.00
				0.51	0.85

STEP 8

Make up table, simular to that shown in Table 1 of this paper. The values for the first 3 columns (sea states 5, 6, and 7) are obtained from the bottom line of step 5, and the values for the "all year average" column, from Step 7.

NOTES:

1. This prediction technique is an approximation which assumes that the ship is always in short-crested (multidirectional) seas, in the North Atlantic, and also that the ship will head equal amounts of time in all directions.
2. No account is taken here of the fact that, to conduct helicopter operations, a ship often has to slow down, particularly in the higher sea states, in order to reduce the relative wind.

APPENDIX 2

DYNAMIC LOADING AND TORQUE PREDICTION METHOD

General

A method for the prediction of Dynamic Loading and Torque for a rigid foil of finite span, undergoing large amplitude pitching, heaving, and surging motions has been derived by Scherer (7). This method has been used successfully to analyze the moments and forces on FF 1052 Class fins (10), and has recently been programmed on FORTRAN by the U.S. Navy. This program (9) also includes a data matrix of test results from a family of low aspect ratio control surfaces.

Theory of Method

Consider a foil with a mean aerodynamic chord of c supported in a stream of velocity U . The foil is performing simple harmonic pitching; heaving, and surging oscillations at a rate of ω radians per second with a mean angle of attack. These motions are defined at a point O located at a distance d ahead of the quarter-chord of the mean aerodynamic chord. Figure A2.1 illustrates the geometry involved. We wish to determine the lift, drag, and moment at O as a function of time. These forces and moments can be separate into contributions from hydrodynamic circulation and added mass. Thus, we have equation (1):

$$L = L_c + L_m \quad (\text{where the subscripts "c" and "m" refer}$$

$$D = D_c + D_m \quad \text{to the contributions from circulation}$$

$$M = M_c + M_m \quad \text{and added mass respectively.)}$$

The oscillating vortex wake downstream of the foil induces a flow at the foil which is a function of the number of chord lengths the foil has travelled in one cycle. This distance can be expressed as π/k where k is the reduced frequency, and is defined as:

$$k = \frac{\omega c/2}{U} \quad \text{equation (2)}$$

This induced flow is characterized by the complex function $c(k)$.

$$c(k) = F(k) + iG(k) \quad \text{equation (3)}$$

(7) References are shown in Appendix and also in main body of paper.

Values of F and G were computed by Theodorsen for infinite span and have since been computed by others for finite span foils and foils under a free surface (Reference (12)). F and G are plotted in Figure A2.2 for infinite span and for elliptic wings of aspect ratio 6, 3 and 0.

As mentioned earlier, this method, as explained completely in reference (11), has been programmed and is used in conjunction with static lift and moment data obtained from reference (4).

Fin Operation

Scherer's method assumes that the fin angle varies sinusoidally with the ship roll angle. In actual practice, improved stabilization can be utilized by using a "bang - bang" control strategy for positioning of the fins. The guidelines for fin operation require that the fins move from hard over to hard over in approximately one sixth of the roll period. Figures A2.3 through A2.6 show the three fin positioning strategies. The "three term Fourier Series" strategy is the one used in the program, as it is considered most representative of real life fin operations.

The equations for the 3-Term Fourier Series are:

$$\alpha = 1.2158 \sin (WT) - 0.2703 \sin (3WT) + 0.0486 \sin (5WT) \quad (4)$$

$$\dot{\alpha} = 1.2158 \cos (WT) - 0.8109 \cos (3WT) + 0.2430 \cos (5WT) \quad (5)$$

$$\ddot{\alpha} = 1.2158 \sin (WT) - 2.4327 \sin (3WT) + 1.1150 \sin (5WT) \quad (6)$$

where α = fin angle, $\dot{\alpha}$ = velocity, $\ddot{\alpha}$ = acceleration

P = amplitude of fin angle oscillation

W = frequency of ship's roll

T = time

Sample Calculation for Patrol Craft

A sample dynamic calculation is contained at the end of this Appendix, so that this method may be used if required. Also the definition of symbols is included for completeness.

Using the method described herein, the normal force and torque were derived for one FFG 7 fin. These are shown in Figure 5 of the main body of this report. Prior to obtaining the values of Figure 5, which are the maximum expected in normal operations, a parametric study was performed, varying the ship speed and fin stock centerline location. Based upon this study, it was determined that a speed of 20 knots would result in the maximum forces and torques, and that the fin stock should be "balanced" at 20% of the mean chord aft of the leading edge.

$U = 16$ fts = 27.024 ft/sec
 (1 Knot = 1.689 ft/sec)
 $T = 6$ sec
 $w = \frac{2\pi}{T} = 1.047 \text{ rad/sec}$
 $k = wc/2U = 0.087$



	t	0	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
u_t	0	0.52	1.05	1.57	2.09	2.62	3.14	3.67	4.19	4.71	5.24	5.76	6.28	
$\sin wt$	0	0.50	0.87	1.00	0.87	0.50	0	-0.29	-0.87	-1.00	-0.87	-0.50	0	
$\cos wt$	1.00	0.87	0.50	0	0.50	-0.87	-1.00	-0.87	-0.50	0	0.50	0.87	1.00	
$h = H \cos wt$	4.00	3.46	2.00	0	-2.00	-3.46	-4.00	-3.46	-2.00	0	2.00	3.46	4.00	
$h = -H \sin wt$	0	-2.09	-3.63	-4.19	-3.63	-2.09	0	2.09	3.63	4.19	3.63	2.09	0	
$\dot{h} = -Hw^2 \cos wt$	-4.39	-3.80	-2.19	0	2.19	3.80	4.39	3.80	2.19	0	-2.19	-3.80	-4.39	
$z = 2 \sin wt$	0	0	0	0	0	0	0	0	0	0	0	0	0	
$\dot{z} = 2w \cos wt$	0	0	0	0	0	0	0	0	0	0	0	0	0	
$\ddot{z} = -2w^2 \sin wt$	0	0	0	0	0	0	0	0	0	0	0	0	0	
$a_1 = 2 \sin wt$	0	0.17	0.29	0.33	0.29	0.17	0	-0.17	-0.29	-0.33	-0.29	-0.17	0	
$a = a_0 + a_1$	0	0.17	0.29	0.33	0.29	0.17	0	-0.17	-0.29	-0.33	-0.29	-0.17	0	
$\sin a$	0	0.17	0.28	0.33	0.28	0.17	0	-0.17	-0.28	-0.33	-0.28	-0.17	0	
$\cos a$	1.00	0.99	0.96	0.95	0.96	0.99	1.00	0.99	0.96	0.95	0.96	0.99	1.00	
$\dot{a} = 2w \cos wt$	0.35	0.30	0.17	0	-0.17	-0.30	-0.35	-0.30	-0.17	0	0.17	0.30	0.35	
$\ddot{a} = -2w^2 \sin wt$	0	-0.18	-0.32	-0.26	-0.32	-0.18	0	0.18	0.32	0.26	0.32	0.18	0	
$A = a_1 g + a_2 a$	2.96	2.30	1.80	1.60	1.80	2.30	2.96	2.30	1.80	1.60	1.80	2.30	2.96	
$P = F(k, A)$	0.99	1.00	1.00	1.00	1.00	1.00	0.99	1.00	1.00	1.00	1.00	1.00	1.00	
$G = G(k, A)$	-0.05	-0.03	-0.02	-0.02	-0.03	-0.05	-0.03	-0.02	-0.02	-0.02	-0.02	-0.03	-0.05	
$Lt = (1.0/w) \arctan (-G/P)$	0.04	0.02	0.02	0.02	0.04	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.04	
$t_2 = t - Lt$	-0.04	0.48	0.98	1.48	1.98	2.48	2.96	3.48	3.93	4.46	4.98	5.48	5.96	
ut_2	-0.43	0.50	1.03	1.55	2.08	2.59	3.10	3.64	4.17	4.69	5.22	5.73	6.23	
$\sin ut_2$	-0.05	0.05	0.86	1.00	0.89	0.52	0.05	-0.48	-0.86	-1.00	-0.88	-0.52	-0.05	
$\cos ut_2$	1.00	0.88	0.52	0.02	-0.48	-0.85	-1.00	-0.88	-0.52	-0.02	0.46	0.88	1.00	
$h_2 = -H_2 \sin ut_2$	0.19	-2.00	-2.59	-4.19	-3.67	-2.19	-0.19	2.00	3.59	4.19	3.67	2.19	0.19	
$\dot{h}_2 = 2w \cos ut_2$	0	0	0	0	0	0	0	0	0	0	0	0	0	

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$a_{11} = p \sin \omega t_1$	-0.02	0.16	0.28	0.33	0.29	0.17	0.02	-0.15	-0.28	-0.23	-0.29	-0.17	-0.02
$a_{12} = a_0 + z_{1i}$	-0.02	0.16	0.28	0.33	0.29	0.17	0.02	-0.15	-0.28	-0.33	-0.29	-0.17	-0.02
$\dot{a}_1 = p_0 \cos \omega t_1$	0.35	0.30	0.18	0.01	-0.17	-0.30	-0.35	-0.31	-0.18	-0.01	0.17	0.30	0.35
$\sin a_1$	-0.02	0.16	0.28	0.33	0.29	0.17	0.02	-0.16	-0.28	-0.33	-0.29	-0.17	-0.02
$\cos a_1$	1.02	0.59	0.96	0.95	0.96	0.99	1.03	0.99	0.96	0.95	0.96	0.99	1.00
$z_1 = (c/2 + d) \dot{z}_1 \cos a_1 - \dot{h}_1$	2.54	2.71	4.00	4.21	3.28	1.48	0.64	-2.73	-4.00	-4.21	-3.28	-1.48	0.64
$E = (c/2 + d) \dot{z}_1 \sin a_1 + U + \dot{L}_1$	27.01	27.14	27.14	27.03	26.91	26.90	27.01	27.14	27.14	27.03	26.91	26.90	27.01
$c' = z_{1i} + \text{erfc}(z_1 \cdot i\pi/2)$	0.004	0.26	0.43	0.49	0.41	0.22	-0.04	-0.26	-0.43	-0.49	-0.41	-0.22	0.004
$z'_c = z^* / \bar{c}^2 + G^2$	0.004	0.26	0.43	0.49	0.41	0.22	-0.04	-0.26	-0.43	-0.49	-0.41	-0.22	0.004
$z_c = a_2 + c'_c$	0.004	0.26	0.43	0.49	0.41	0.22	-0.04	-0.26	-0.43	-0.49	-0.41	-0.22	0.004
$Y = \text{erfc}(z_1 \cdot i\pi(c/2)/(U + \dot{L}))$	0.03	0.02	0.01	0	-0.01	-0.02	-0.03	-0.02	-0.01	0	0.01	0.02	0.03
$L_3 = z_1 - z_2 + Y(1.0 - Y/4a'_c)$	-0.03	-0.07	-0.13	-0.16	-0.13	-0.07	0.03	0.07	0.13	0.16	0.13	0.07	-0.03
$\sin L_2$	-0.03	-0.07	-0.13	-0.16	-0.13	-0.07	0.03	0.07	0.13	0.16	0.13	0.07	-0.03
$\cos L_2$	2.00	1.00	0.99	0.99	1.00	1.00	1.00	1.00	0.99	0.99	0.99	1.00	1.00
$y_1 = (c/e + d) \dot{z} \sin a + U + \dot{L}$	27.02	27.09	27.09	27.02	26.96	26.97	27.02	27.09	27.09	27.02	26.96	26.97	27.02
$y_2 = (c/e + d) \dot{z} \cos a - \dot{h}$	0.43	2.45	3.84	4.19	3.42	1.72	-0.43	-2.46	-3.84	-4.19	-3.42	-1.71	0.43
$v = v_1^2 + v_2^2$	27.03	27.20	27.35	27.35	27.16	27.02	27.02	27.19	27.38	27.35	27.16	27.03	27.02
C_2	0.036	0.621	0.766	0.782	0.759	0.583	-0.015	-0.621	-0.766	-0.782	-0.759	-0.583	0.036
C_3	0.012	0.052	0.182	0.210	0.178	0.059	0.012	0.062	0.182	0.216	0.178	0.059	0.012
C_N	0.010	0.037	0.016	0.011	0.018	0.039	-0.010	-0.037	-0.016	-0.011	-0.018	-0.039	0.010
$L_1 = C_1 0.5 \delta^{1/2} S$	0.79	13.72	17.11	17.40	16.74	12.71	-0.79	-13.72	-17.11	-17.40	-16.74	-12.71	0.79
$L_2 = C_2 0.5 \delta^{1/2} S$	0.26	1.37	4.23	4.83	3.93	1.29	0.26	1.37	4.23	4.83	3.93	1.29	0.26
$L_3 = C_3 0.5 \delta^{1/2} S$	0.93	3.68	1.61	1.11	1.79	3.83	-0.99	-3.68	-1.61	-1.11	-1.79	-3.83	0.93
$\tilde{h} = [L + h] \sin a + [(U + L)\dot{a} - h] \cos a + [c/4 + d]\dot{a}$	13.77	11.46	6.58	-0.41	-6.95	-0.06	-13.77	-11.46	-6.58	-0.41	6.95	13.77	6.06

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$\sin \alpha$	0	0.26	0.42	0.47	0.40	0.22	0	-0.26	-0.42	-0.47	-0.40	-0.2	0
$\cos \alpha$	1.00	0.97	0.91	0.88	0.92	0.98	1.00	0.97	0.91	0.83	0.92	0.98	1.00
$\dot{v}_C = (\dot{L} + \dot{h}\dot{a}) \cos \alpha - [(\dot{U} + \dot{D})\dot{a}$ $- h] \sin \alpha$	0	-2.59	-2.56	0	2.56	2.59	0	-2.59	-2.56	0	2.56	2.59	0
$\Sigma = A_{27} (c/2)^2 b / \sqrt{1.0 + \dot{A}^2}$	0.20	0.19	0.18	0.18	0.19	0.20	0.19	0.18	0.18	0.18	0.19	0.19	0.20
$\kappa = 0.33 + 0.076 A - 0.005 A^2$	0.43	0.42	0.41	0.40	0.41	0.42	0.43	0.42	0.41	0.40	0.41	0.42	0.43
$L_2 = \Sigma \dot{v}_N$	2.75	2.29	1.32	-0.17	-1.39	-1.21	-2.75	-2.29	-1.32	0.17	1.39	1.21	2.75
$D_2 = \Sigma (c/2)^2 \dot{v}_C$	0	-0.01	-0.01	0	0.01	0	0.01	-0.01	0	0.01	0	0.01	0
$v_2 = -\Sigma [(c/4) (\dot{U} + \dot{L})\dot{a} \cos \alpha + \kappa (c/4)^2 \dot{a}]$ $- (2.1)] - 1.79$	-0.98	0.39	1.05	1.82	2.11	1.71	0.98	-0.04	-1.05	-1.82	-2.11		
$\alpha_j = \alpha - L_2$	0.02	0.24	0.42	0.49	0.42	0.24	-0.03	-0.24	-0.43	-0.49	-0.42	-0.24	0.03
$\sin \alpha_j$	0.03	0.24	0.41	0.47	0.41	0.24	-0.03	-0.24	-0.41	-0.47	-0.41	-0.24	0.03
$\cos \alpha_j$	1.00	0.97	0.91	0.88	0.91	0.97	1.00	0.97	0.91	0.85	0.91	0.97	1.00
$L_C = L_1 \cos (\dot{L}_2) - D_1^2 \sin (\dot{L}_2)$	0.80	13.82	17.48	18.08	17.08	12.80	-0.80	-13.82	-17.48	-18.08	-17.08	-12.80	0.80
$D_C = D_1 \cos (\dot{L}_2) + L_1 \sin (\dot{L}_2)$	0.24	0.41	1.93	1.98	1.71	0.40	0.24	0.41	1.93	1.96	1.71	0.40	0.24
$\kappa_C = \kappa_1 - \dot{d} [L_1 \cos \alpha_j + D_1 \sin \alpha_j]$	11.87	23.70	-6.62	-13.16	-3.79	27.00	-11.87	-23.70	6.62	13.16	3.79	-2.70	11.87
$L_1 = L_2 \cos \alpha - D_2 \sin \alpha$	2.72	2.26	1.27	-0.16	-1.34	-1.20	-2.75	-2.26	-1.27	0.16	1.34	1.20	2.75
$D_1 = D_2 \cos \alpha + L_2 \sin \alpha$	0	0.36	0.36	0	-0.36	-0.36	0	0.36	0.36	0	-0.36	0	0.24
$\dot{v}_1 = v_2 - (c/4 + \dot{d}) L_2$	-56.6	-55.8	-31.6	3.0	33.5	40.0	66.6	55.8	31.6	-5.0	-33.5	-40.0	-56.6
$L = L_C + L_H$	3.6	16.1	18.8	17.9	15.7	11.6	-3.6	-16.1	-18.8	-17.9	-15.7	-11.6	3.6
$D = D_C + D_H$	0.2	0.8	2.3	2.0	1.4	0	0.2	0.8	2.3	2.0	1.4	0	0.2
$\kappa = \kappa_C + \kappa_H = Q_H$	-54.7	-32.1	-38.2	-10.2	29.7	67.0	54.7	32.1	38.2	10.2	-39.7	-57.0	-51.7
$F_R = \sqrt{L^2 + D^2}$	3.6	16.1	18.9	18.0	15.8	11.6	3.6	16.1	18.9	18.0	15.8	11.6	3.6
$F_H = L \cos \alpha_C + D \sin \alpha_C$	3.6	15.9	18.1	17.1	15.2	11.5	3.6	15.9	18.1	17.1	15.2	11.5	3.6
$Q_Z = 0.03 c F_H$	5.8	25.8	29.3	27.7	24.6	18.6	5.8	25.8	29.3	27.7	24.6	18.6	5.8
$Q_T = 3 F_R$	10.1	45.2	53.1	50.6	44.4	32.6	10.1	45.2	53.1	50.6	44.4	32.6	10.1
$Q_H + Q_Z$	-48.9	-6.3	-8.9	17.5	54.3	85.6	60.5	57.9	67.5	37.9	-5.1	-43.4	-46.9
$Q_H - Q_E$	-60.5	-57.9	-67.5	-37.9	5.1	48.4	48.9	6.3	8.9	-17.5	-53.3	-85.6	-60.5
$Q_T + Q_Z$	-38.8	38.9	44.2	68.1	98.7	118.2	70.6	103.1	120.6	88.5	39.3	-15.8	-38.8
$Q_H - Q_T$	-70.6	-103.1	-120.6	-88.5	-39.3	15.8	38.8	-38.9	-44.2	-68.1	-98.7	-113.2	-70.6

DEFINITIONS OF SYMBOLS

A	Effective aspect ratio
A_g	Geometric aspect ratio
b	Mean span of fin
B	Ship beam
c	Mean chord of fin
\overline{CP}	Center of pressure
$C_{\overline{CP}}$	Center of pressure coefficient
C_D	Drag coefficient
C_L	Lift coefficient
C_M	Moment coefficient
C_N	Normal force coefficient
C_R	Resultant force coefficient
d	Distance from quarter-chord to stock
D	Total drag
D_C	Drag due to circulation
D_m	Drag due to added mass
D_1	Steady state drag
D_2	Chordwise force from added mass
E	Coefficient for determination of β ,

$$E = \frac{\lambda_U}{\lambda_U - \lambda_L}$$

F	Real part of complex variable describing vortex wake
F_N	Normal force
F_R	Resultant force
G	Imaginary part of complex variable describing vortex wake
h	Heave displacement of fin at time t
h_i	Heave displacement of fin at time t_i
H	Amplitude of sinusoidally-varying heave of fin
k	Reduced frequency of fin oscillation
K	Constant related to radius of gyration of added mass
ℓ	Surge displacement of fin at time t
ℓ_i	Surge displacement of fin at time t_i
L	Total lift
L_c	Lift due to circulation
L_m	Lift due to added mass
L_1	Steady state lift
L_2	Force from added mass normal to chord
m	Added mass
M	Total moment at stock
M_b	Maximum bending moment of stock
M_c	Moment at stock due to circulation
M_m	Moment at stock due to added mass
M_1	Steady state moment at quarter-chord

M_2	Mid-chord moment due to added mass
P	Amplitude of fin angle oscillation
Q	Maximum absolute value of either the upsetting torque $Q_H + Q_E + Q_F$ or the restoring torque $Q_H - Q_E - Q_F$, whichever is larger (torsion to stock)
Q_E	Torque (at stock) allowance for error in center of pressure determination
Q_F	Torque to overcome friction in bearings
Q_H	Hydrodynamic torque at stock
R	Ship roll amplitude
R_L	Radius of the lower bearing (stock radius plus the thickness of liner or radius to the center of the bearing rollers or balls)
R_U	Radius of the upper bearing
S	Fin area
t	Time
t_i	Time preceding t by increment Δt
T	Period of ship roll
U	Ship speed
V	Effective flow velocity considering ship roll
v_1	Instantaneous horizontal velocity of fin
v_2, v_R	Instantaneous vertical velocity of fin due to roll
\dot{w}_C	Chordwise acceleration of water relative to mid-chord

\dot{w}_N	Acceleration of water in direction to normal to chord relative to mid-chord
z	Amplitude of sinusoidally-varying surge
α	Fin angle
α_e	Effective angle of attack
α'_e	Contribution to effective angle of attack due to fin oscillation
α_o	Center of fin oscillation
α_l	Oscillatory portion of fin angle at time t
α'	Instantaneous angle of attack at 3/4 chord evaluated at time t_i
α_i	Fin angle at time t_i
α_j	$\alpha - \Delta\alpha$
α_{li}	Oscillatory portion of fin angle at time t_i
β	Bearing friction factor ($Q_F = \beta F_R$)
γ	Angle between horizontal and vertical velocity vectors
δ	Distance from leading edge to stock
$\Delta\alpha$	Inclination of force vectors due to circulation
θ	Instantaneous ship roll angle
λ_F	Distance from ship's \mathbb{E} through CG to mean chord of the fin
λ_L	Distance from mean chord to the lower stock bearing
λ_U	Distance from mean chord to the upper stock bearing

M_L	Coefficient of friction of lower bearing
M_U	Coefficient of friction of upper bearing
ρ	Water density
σ	Additional angle of attack in steady state due to ship roll
τ	Mean thickness of chord
ψ	Stock penetration into hub
ω	Frequency of ship roll and fin angle oscillations
Ω	Sweep angle

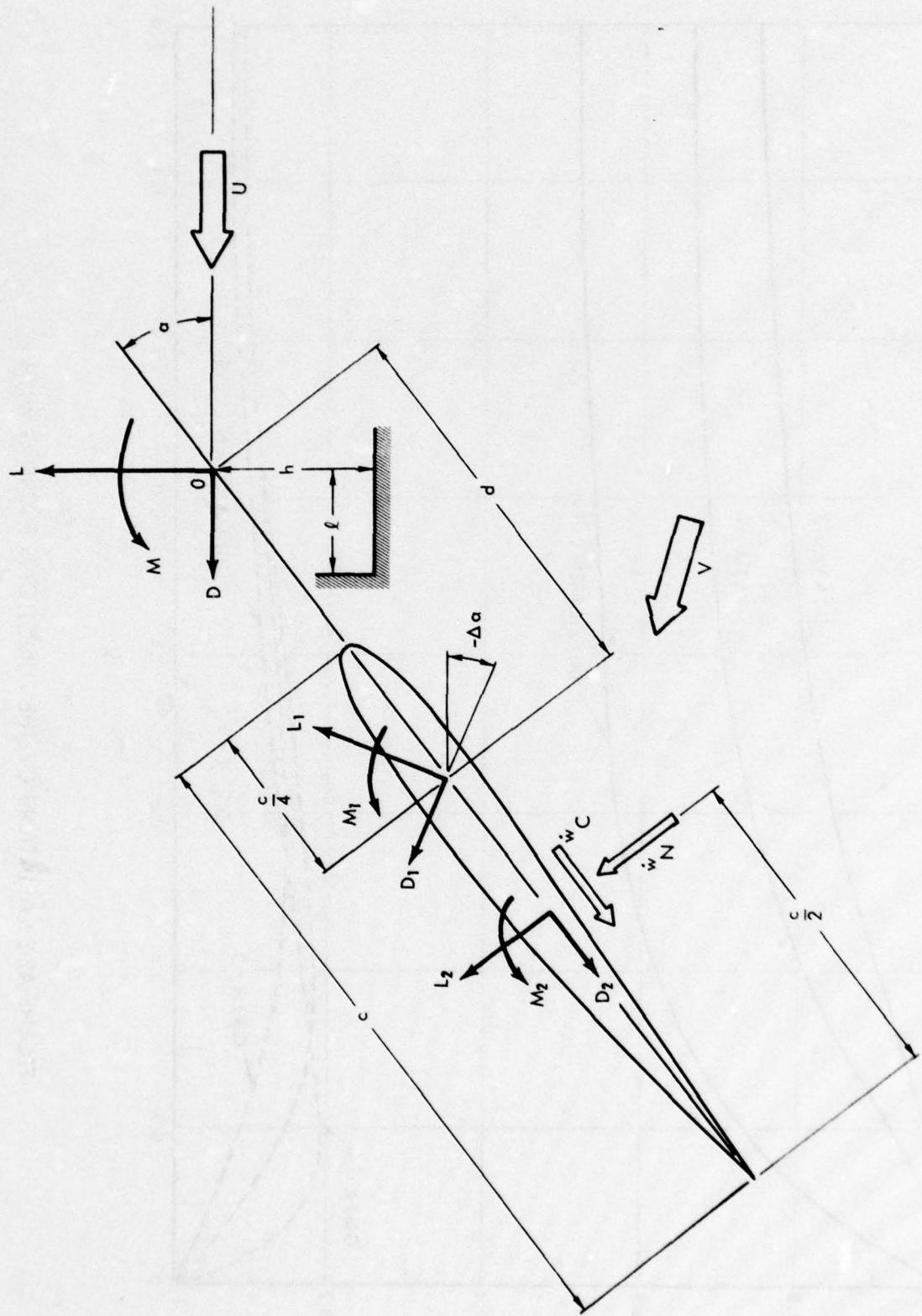


FIGURE A2-1 DEFINITION SKETCH OF FORCES AND MOMENTS OF FOIL

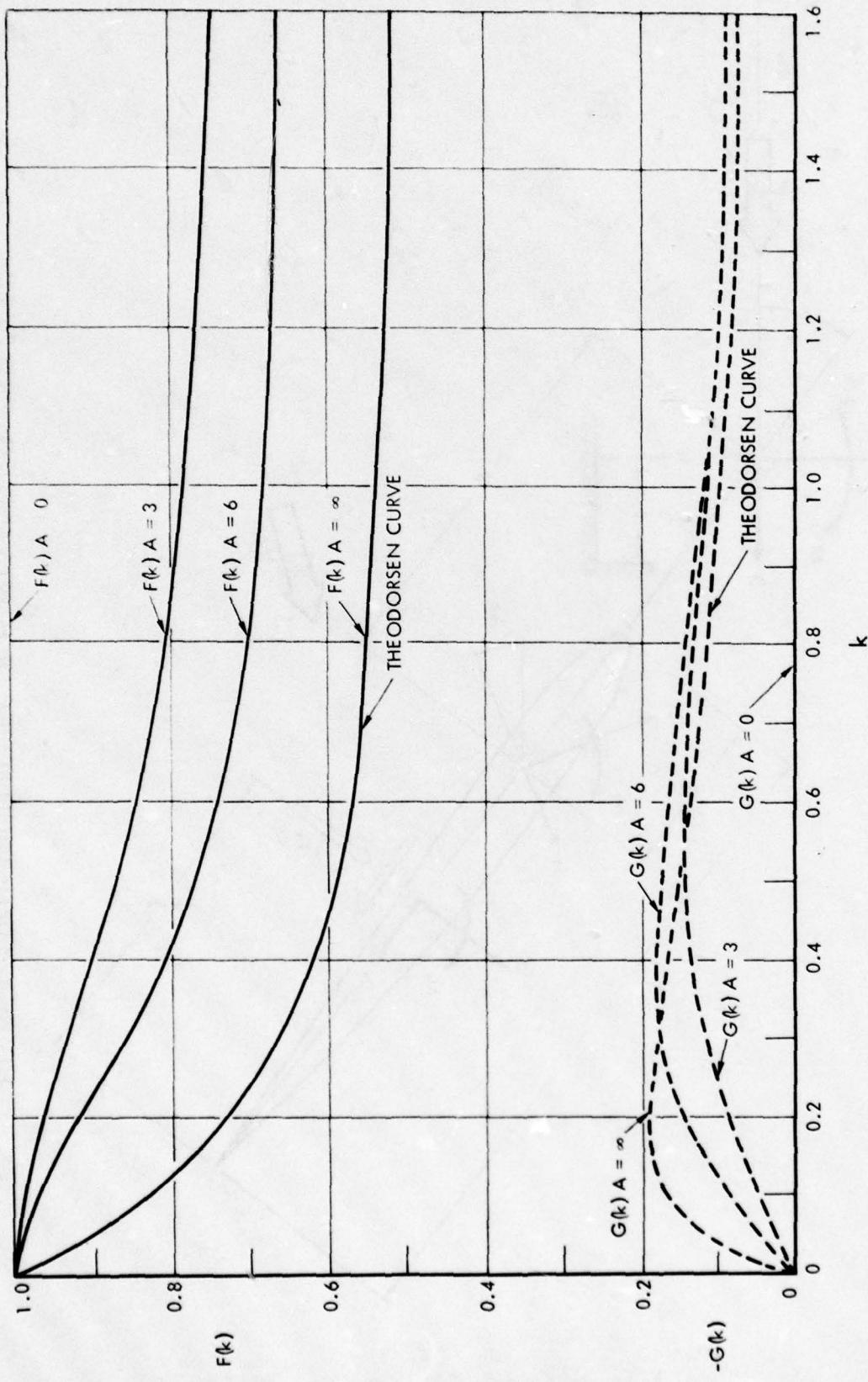
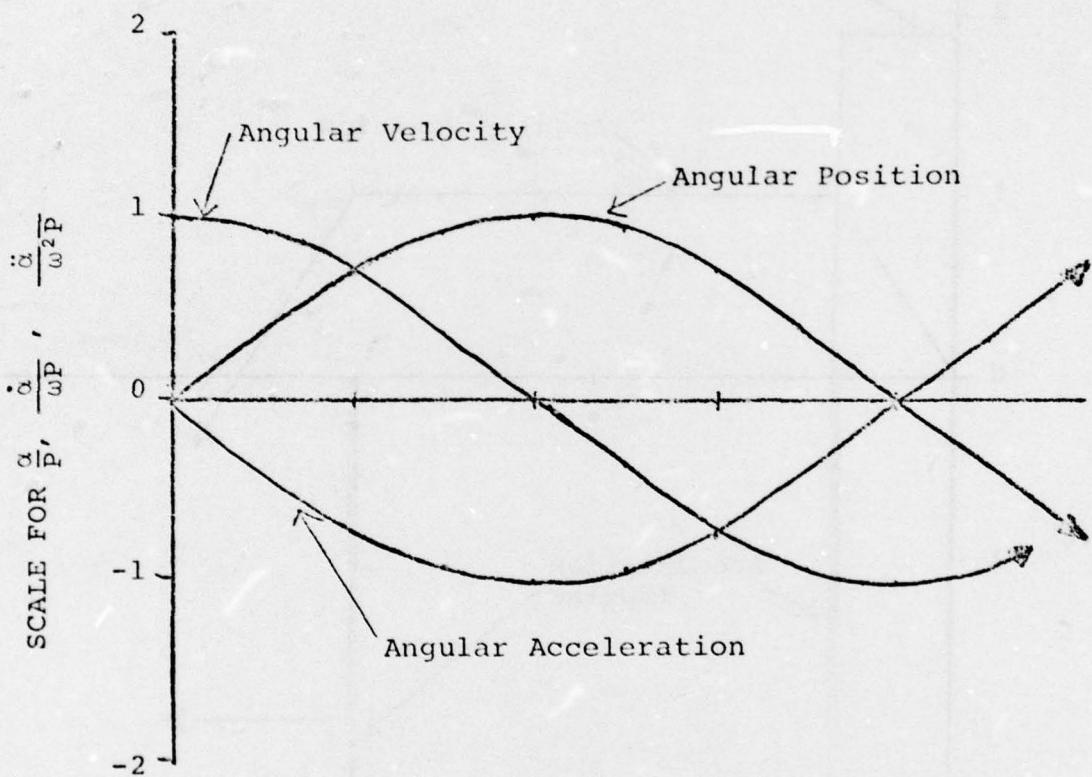
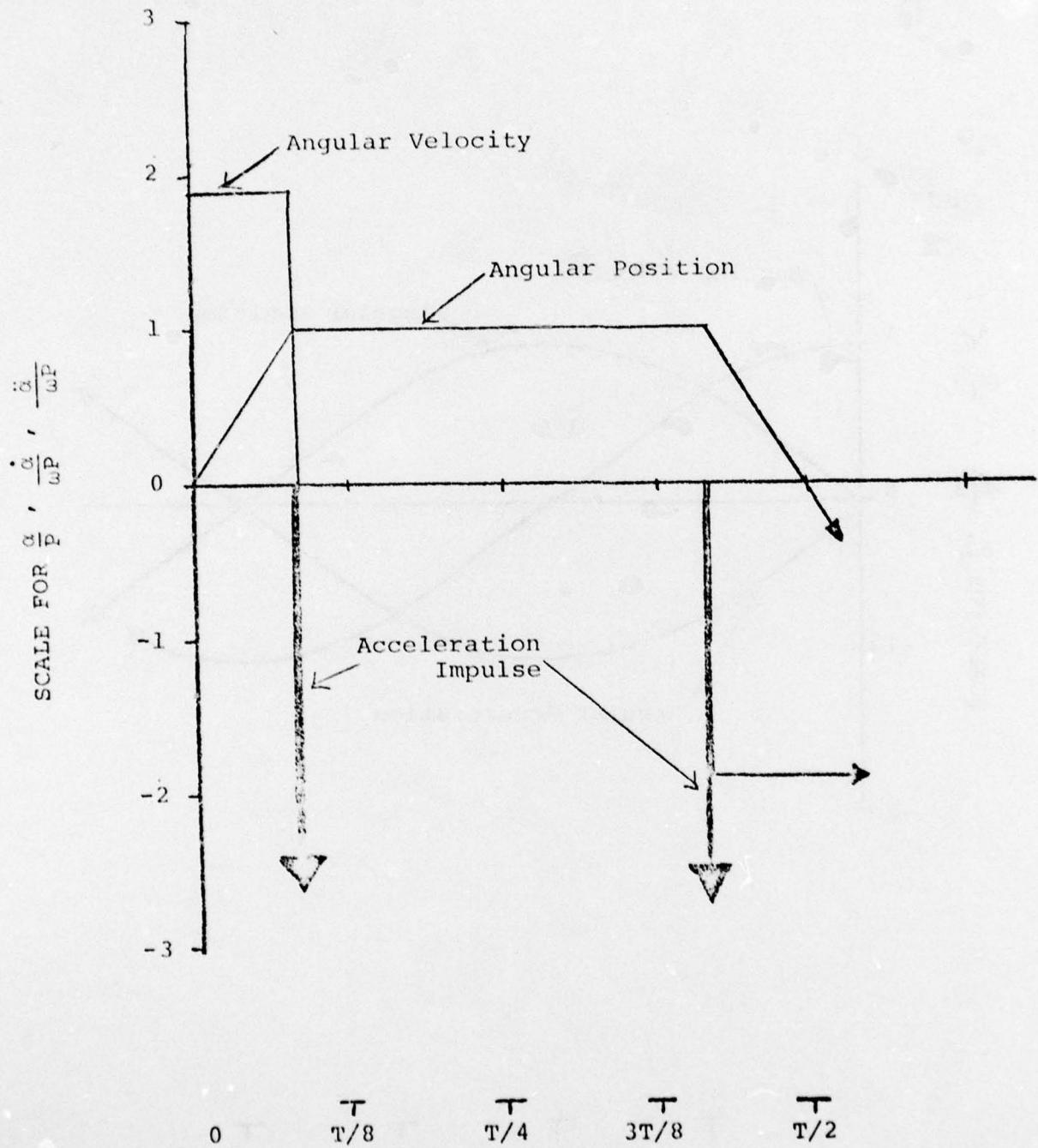


FIGURE A2-2 VARIATION OF THE FUNCTIONS F AND G WITH k



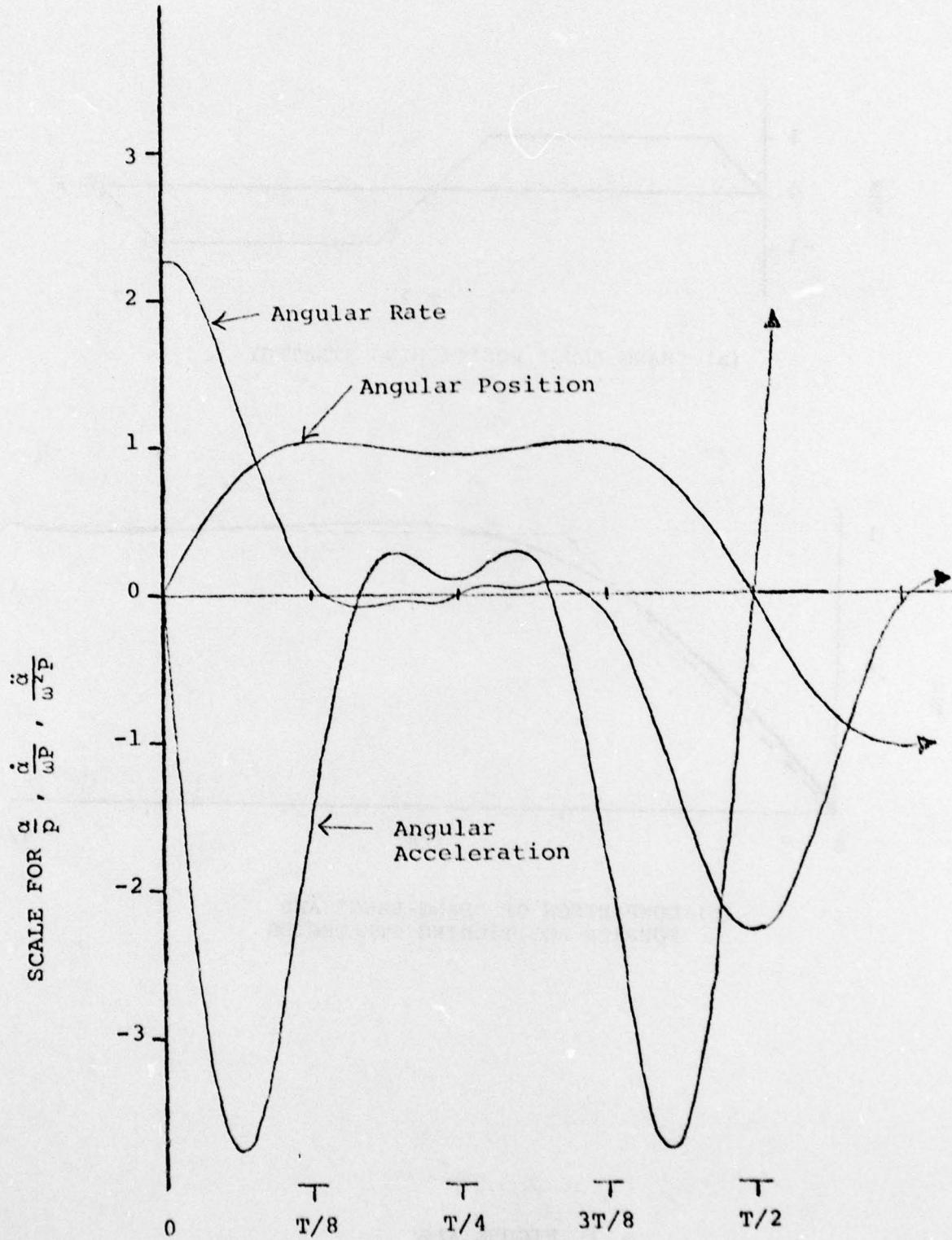
SINUSOIDAL POSITIONING STRATEGY

FIGURE A2-3

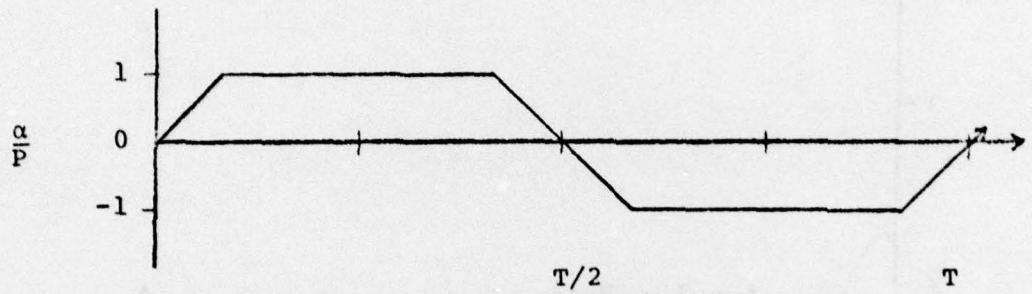


"BANG-BANG" POSITIONING STRATEGY

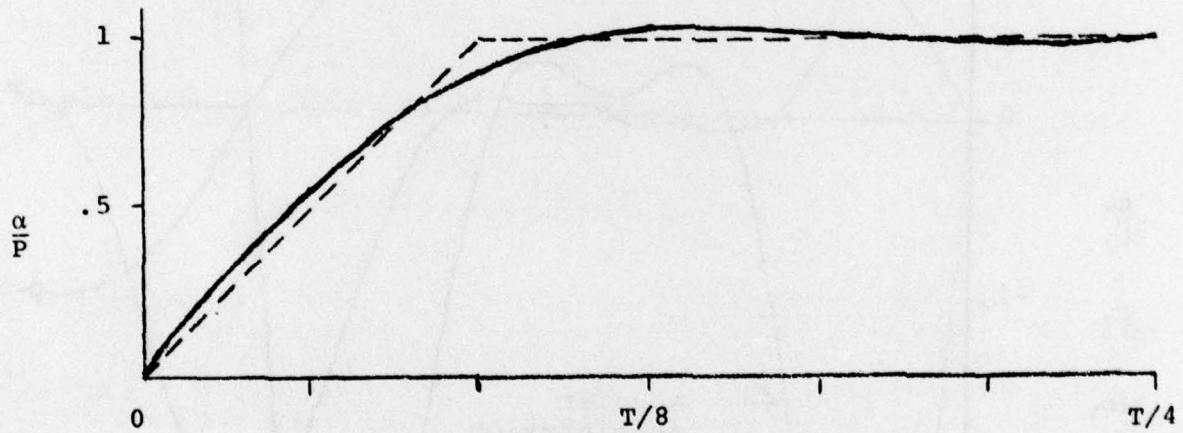
FIGURE A2-4



THREE TERM FOURIER SERIES POSITIONING STRATEGY
FIGURE A2-5



(a) "BANG-BANG" POSITIONING STRATEGY



(b) COMPARISON OF "BANG-BANG" AND
FOURIER POSITIONING STRATEGIES

APPENDIX 3
FIN STABILIZATION SYSTEM
SPECIFICATION DETAILS

1. Fin Subsystem

a. Fins - The fins will have streamlined NACA 0015 cross-sections. Tip fairing will be the 0015 half thickness rotated to be half a body of revolution. Fin planform and fin stock location will be as given in Figure 6. Large fillets will be used in those parts of the fin structure where sharp corners might cause harmful stress concentrations. The fairness and smoothness of the fin surface will be such as to minimize cavitation and turbulent flow generation. To achieve this, special precautions will be taken, as follows: All external welds will be ground smooth and flush with the contour. No weld spatter, pits, or other roughness will be permitted. Weld undercuts and other sharp surface deviations will be welded and ground flush. Fin stocks and fin hubs will have machined tapers or good engagement and to facilitate unshipping.

b. Fin Stocks - The fin stock will be designed to accommodate the hydrodynamic loads imposed on it by the fin and the dynamic torque loads imposed on it by the actuators. To minimize weight, and to provide for air supply, the fin stock will be hollow. A protective sleeve of gunmetal or monel will be shrunk fit on the fin stock in way of the outboard sleeve bearing and its upper and lower packings and seals, and will extend to the root of the fin.

c. Fin Stock Bearing Assemblies - Each fin stock will be provided with an inboard and outboard bearing assembly and associated grease supply systems. The inboard bearing will be designed to carry both radian and thrust loads imposed on it by the fin hydraulic actuators as well as the weight of the components being supported by the fin stock. For the inboard bearing of each fin shaft a spherical roller bearing per FF-B-185 will be used. The outboard bearing will be designed for radial loading only. Laminated phenolic stave type bearings will be used for the outboard fin stock bearings. All bearings will be fitted for pressure grease fittings. Positive means will be provided to prevent bushings from turning and cutting off the supply of lubricant.

d. Fin Stock Seals and Packings. Approximate packings and seals will be provided in way of each outboard bearing. The inboard hull packing will be located above the outboard fin stock bearing. Space will be provided for a stuffing box and 7/8 inch square or larger packing material assembled in the following order (starting outboard and working inboard):

- 1st - One ring neoprene and cotton duck
- 2nd - One ring flax
- 3rd - One ring flax
- 4th - Latern grease ring
- 5th - One ring flax
- 6th - One ring neoprene and cotton duck

These packing materials are preferred, but the contractor may propose alternatives. The outboard hull seal will be located below the outboard fin stock bearing and will be composed of one or more whole (not split) spring loaded lip seal(s) with the lips of the seal(s) facing outboard. Hull seals, glands, retainer rings, stuffing boxes, compression packing, and gland follower rings adjacent to the fin stock will be designed to accommodate fin stock deflections due to bending moment and maximum bearing clearance for all operating conditions.

e. Air Emission System - An air emission system suitable for future completion will be provided for each fin subsystem. The contractor will provide separately controlled air supply lines for the leading edge and tip of the fin. The air supply line for the leading edge will supply a manifold suitable for accepting the later installation of nozzles located up to 30 degrees above and below the section chord (when measured about the center of curvature of the section at the nose) and spaced on about 6 inch centers. The air supply line for the tip will supply a similar manifold for the later installation of nozzles located as along the leading edge, except on about 3 inch centers.

f. Tiller Assembly - A tiller assembly will be provided for each fin stock. Each tiller assembly will include two arms mounted on opposite sides of a split tiller hub. The hub will be keyed to the fin stock. The tiller hub will include provisions to hit steel hard stops one degree prior to bottoming out of the hydraulic cylinders, CW and CCW, without resulting in any interference between the tiller assembly and the tilting cylinders. The tillers and the hardstops shall be designed so that a steel block may be inserted between the two for test purposes.

g. Grease Supply Systems - Each outboard bearing assembly will have an independent grease supply system to supply grease to the laminated phenolic bearings, and the latern grease ring. Each grease supply system will consist of a hand operated grease pump, a pressure gage, a manifold with at least eight hand operated distribution valves, and a total of eight CRES grease lines to the following lubrication points: four uniformly distributed points on the circumference of the laminated phenolic bearing and four uniformly distributed points on the circumference of the latern grease ring. The port and starboard grease pumps will be located in accessible locations. Grooves will be provided in the phenolic bearing assembly to evenly distribute the grease from the four lubrication

points to the working surfaces of the bearing. The spherical roller bearings will be fitted with pressure grease fittings for greasing with a portable grease gun. Grease lubrication vents for the roller bearing will be distributed in 90 degree intervals and be routed out through the bearing housing individually.

h. Support Structure and Mounting Bedplate - The fin subsystem will be designed for installation from the exterior of the ship, utilizing a standardized bolted foundation ring. However, subject to Navy approval, the contractor may deviate from the above in order to optimize the fin subsystem and hydraulic subsystem arrangement.

2. Hydraulic Subsystem - An independent hydraulic subsystem will be provided for each fin subsystem. Each hydraulic subsystem will be arranged functionally in accordance with Figure 7A. The design of the subsystem will be such that the hydraulic resistance of the variable delivery pump/actuator loop (pilot operated bypass valve, piping, fittings, manifolds, hoses) will be kept to a minimum. Air vents will be provided at all points in the hydraulic subsystem where air will tend to accumulate during shipboard operations. Subsystem cleanliness will be maintained by a Navy approved method in accordance with MIL-STD-419. Cleaning by mechanical means, followed by dry air blast will be used where cleaning of pumps, motors, rams, valves, and other accessories becomes necessary.

a. Main Hydraulic Pumps - The hydraulic pumps will be of the reversible variable delivery piston type, and will conform to MIL-P-17869. To preclude cavitation the system will provide for replenishing the pump by means of a charge pump, an accumulator and shuttle valve. The nominal pump speed will be 1200 rpm or less.

b. Charge Pumps - Each main hydraulic pump will be supplied with a charge pump. The charge pumps will be directly driven by the variable delivery pump drive shafts and will be manifold mounted on the outside of the variable delivery pump housings.

c. Charge Pump Relief Valve - Each charge pump will be provided with a manifold type relief valve.

d. Charge Pump Replenishment Check Valve - The charge pump will connect with the variable delivery pump output lines via spring loaded manifold type check valves.

e. Pump Stroking Control Assembly - The variable delivery pump swashplate stroking control will include a subplate mounted electro-hydraulic servovalve with MS type electrical connector, a manifold assembly with ports for an external servo valve supply line filter, and a precision servo potentiometer conforming to MIL-R-39023. The potentiometer will be mechanically coupled to the pump swashplate to sense swashplate

displacement. The potentiometer and servo valve will be independent of one another and it will be easy for Ship's Force to replace either one without disturbing the other. Adapter manifolds will be provided as required to allow servovalves from at least two independent commercial sources to be used interchangeably.

f. Electric Motors and Motor Controllers - Motors for the hydraulic power units will be furnished in accordance with MIL-M-17060 and motor controllers in accordance with MIL-C-2212 will be used for motor starting and stopping, and control of the solenoid actuated pilot valve that controls the flow of hydraulic oil to the pump bypass/cylinder blocking (PBCB) valve that is associated with that particular pump and motor combination.

g. Pump-Motor Coupling - Each electric motor drive shaft will be connected to its hydraulic pump drive shaft via a spacer type flexible coupling so that it shall be possible to replace pump drive shaft seals without removing the pump or motor.

h. Pump-Motor Mounting - The electric motor(s), main hydraulic pump(s) and other hydraulic components where practicable will be mounted on a common support structure. Flange mounting of the pumps and motors to permit removal and assembly with minimum alignment problems is preferred; however, the use of other configurations in order to optimize arrangements is acceptable. To reduce noise transmission, the support structure shall be mounted on the suitable resilient mountings in accordance with MIL-M-17508.

i. Acoustic Filters - In each hydraulic subsystem, the hydraulic fluid flowing to the tilting cylinders passes through an acoustic filter.

j. Relief Valve Manifold - One relief valve manifold will be provided for each variable delivery pump and will comprise two cross line relief valves, a shuttle valve, and a charge pump relief valve.

k. Hydraulic Actuators - A tilting cylinder assembly will be provided for each fin stock. When activated, this assembly will apply the required force to the tiller assembly for setting fin angle of attack. Two tilting cylinders will be provided for each assembly. The cylinders will be commercially available types suited for at least 3000 psi working and 4500 psi test pressures. The tilting cylinders will incorporate a cushioning action at each end of maximum piston travel to smoothly decelerate and stop the fins when the limits of the travel are reached. Standards, commercially available "U" or "V" ring type piston and rod seals will be provided. Spherical ball bushings will be used at the attachment points of each cylinder. These bushings will comprise a spherical ball within an outer spherical housing. The bushings shall be provided with seals and will be pressure grease lubricated.

1. Grease Supply System - A grease supply system will be provided for the spherical ball bushings. The system will consist of a hand-operated pump, a grease container, a pressure gage, a manifold with four hand-operated supply valves and grease lines to the lubrication points.

m. Tanks - A supply tank and an auxiliary head tank will be provided for each hydraulic subsystem. The supply tank in each hydraulic subsystem will provide hydraulic fluid cooling. Each supply tank will be designed to hold at least 40 gallons of hydraulic fluid and will be designed for installation such that one side is made up of the ship's hull, exposed to a free flow of ambient temperature sea water. The sides of the tank to be welded to the ship's hull will include at least one inch extra steel for trimming by the shipbuilder. Baffles will be provided that cause the inlet hydraulic fluid to move in a back and forth fashion across the side of the tank exposed to sea water. The baffles will be readily removable so that thorough cleaning of the tank interior may be accomplished with minimum risk of trapped contaminants or uncleared areas. The auxiliary head tank will be fabricated from CRES, and will be designed for the installation into space provided directly below the second deck. The auxiliary head tank will be provided with a 10 micron nominal air breather filter. The breather will be equipped with a dual air relief to maintain a static pressure head tank from 0.5 psi vacuum to 3 psi positive pressure, at the top of the auxiliary head tank. No breathing will occur within these limits.

n. Pump Bypass/Cylinder Blocking Valve (PBCB) - A PBCB valve will be provided that connects the output of each variable delivery pump to the tilting cylinders when it is activated, and which bypasses the pump and simultaneously blocks the valve cylinder ports, preventing fluid from entering or leaving the tilting cylinders (hydraulic lock) when it is deactivated. The PBCB valve will be a manifold mounted, 4-way, open center directional control valve fitted with a hydraulic actuator for remote operation.

o. PBCB Valve, Pilot Valve, and Relief Valve Manifold - The PBCB valve, the pilot valve and the variable delivery pump relief valve manifold for each variable delivery pump will all be mounted on a common manifold provided for each pump. Each manifold will be identical.

p. Hydraulic Filters - Filters will be readily accessible and all filter elements shall be removable for service and inspection without disconnecting the attached pipe or dismounting the filter housing. Frequency of servicing shall be indicated on a label plate attached to the filter. All filters shall have a differential pressure indicator to show when the filter needs replacement. Filler pipes for tanks shall include a removable filter screen of 180 mesh or finer. Breathing pipe caps shall have cleanable, 10 micro (nominal) or finer, air filters. Identical filter assemblies shall be installed in the replenishment

return line and the servo valve supply line. Filter assemblies will use disposable filter elements in accordance with MIL-F-24402. The servo valve supply filter will be installed between the charge pump and servo valve supply port using the manifold porting available in the servo valve controller assembly.

The replenishment return line filter shall be installed between the variable delivery pump case drain output connection and the supply tank. The return line filter shall include a bypass relief valve. Means shall be provided to change filter elements without the requirement for draining the auxiliary head or supply tank. Filter elements shall be capable of being replaced with minimal fluid spillage. An instruction plate shall be provided in the vicinity of the filters that provides instructions on replacement of the filter elements.

q. Accumulators - A bladder type accumulator shall be provided for each charge pump to supplement the pump when it is supplying fluid for replenishment, for actuating the Pump Bypass/Cylinder Blocking (PBCB) valve, and for supplying the servo valve.

r. Fin Centering Pump - A hand operated pump will be provided for hand positioning, or emergency centering of the fin.

s. Charge Pump Self Test Circuit - A manually operated open center, directional control valve will be provided between the replenishment return line filters and the supply tanks to selectively direct the return flow of any charge pump through a calibrated orifice for monitoring the charge pump flow rate.

t. Miscellaneous Valves - Wherever stop valves are employed, protection will be provided against the inadvertent application of destructive pressure when the stop valve is closed. Provision will be made for locking adjustable valves at service adjustment to prevent tampering. Valve operation will be such as to prevent detrimental surges in the hydraulic subsystem.

u. Tilting Cylinder Stop and Bypass Valves - Each tilting cylinder line will be provide with a manually operated stop valve. A full flow manually operated valve will be provided in the tilting cylinder piping to bypass the tilting cylinders.

v. Sampling Valves - Hydraulic fluid sampling valves will be provided for taking samples from the high pressure lines between the variables delivery pumps and the hydraulic manifolds.

w. Cooling Control Valves - A manually operated directional control valve will be provided that can divide the tank inlet fluid between two ports on the tank. One port will route the inlet fluid into the baffles that cause the fluid to move adjacent to the hull plating for maximum cooling. The other port will bypass the inlet fluid around the baffles so that there is little cooling.

x. Fill and Drain Pump - A hand operated fill and drain pump will be provided on each subsystem.

y. Hydraulic Fluid Hydraulic fluid will be petroleum base of a type or grade selected from MIL-L-17672.

3. Control Subsystem

a. Subsystem Definition - The control subsystem will provide signals to the hydraulic subsystem, monitor critical system parameters, and enable manual positioning and control of fin operation. Control subsystem interfaces with the hydraulic subsystem, the fin subsystem, the EM log and ship's gyro are indicated in Figure 9. The control subsystem will consist of eight electronic units as shown in Figure 10. The control subsystem will consist of the following individually enclosed units:

- (1) Bridge Control Unit (BCU)
- (2) Central Control Unit (CUU)
- (3) Central Processor Unit (CPU)
- (4) Auxiliary Sensor Unit (ASU)
- (5) Local Control Unit (LCU) Port
- (6) Local Control Unit (LCU) Starboard
- (7) Angle Transmitter Unit (ATU) Port
- (8) Angle Transmitter Unit (ATU) Starbord

b. General Requirements - The control subsystem will be designed in accordance with the requirements of MIL-E-16400.

c. Requirements for Standard Electronic Modules (SEM) - Electronic controls will be designed using Navy Standard Electronic Module Program (SEMP) modules in accordance with MIL-STD-1378.

(1) SEMP Deviation - SEMP will be implemented in all electronics except:

- (a) Bridge Control Unit
- (b) Central Control Unit
- (c) Power Supplies (and power supply load resistors if required)
- (d) Fuses
- (e) Off/On/Remote Switch in the Central Processor Unit (CPU)
- (f) Switches, Indicators, and Jacks in the Local Control Unit
- (g) Solid state relays in motor controllers, which are part of the hydraulic subsystem.

(2) Development of New Modules - In the event appropriate SEMP designs are unavailable, MIL-M-28787 and MIL-STD-1389 will be used as the development document for new designs. All SEMP standard and special modules are to be documented in accordance with MIL-STD-1378, Appendix A, Type C2a specifications. The module specification will be subject to review by the SEMP-QA activity.

(3) Module Environment Requirements - All modules proposed for use in the fin stabilizer system will conform to the Class I environmental requirements of MIL-STD-1389.

d. Dynamic Computer Simulation - Engineering estimates regarding system stability and performance will be validated by means of dynamic computer simulation tests, utilizing an analog computer, its digital equivalent, or a hybrid computer. The machinery and controls simulations will include all significant non-linearities such as dead zone, backlash, hysteresis, and saturation in the servo valve and pump stroke mechanisms.

e. Bridge Control Unit (BCU) - A guidance configuration of the BCU is shown in Figure 11. The BCU will be panel mounted in the ship's control console, on the bridge of the ship. All indicator lamps in the BCU will receive power from the 6 volt AC master dimmer in the ship's bridge control console.

f. Central Control Unit (CCU) - A guidance configuration of the CCU is shown as Figure 12. The CCU will be the master controller for the fin stabilizer system, and will consist of two major subassemblies, the Central Panel Assembly (CPA), and the Operator Control Module (OCM), which will plug into the CPA. The CCU will be located in the ship's central control station. The OCM will be mechanically and electrically interchangeable with the Bridge Control Unit (BCU), except that the OCM will have additional operational controls and indicators. The CPA will alternatively accept the Operator Control Module (OCM) or the Bridge Control Unit (BCU) as plug-in modules. A panel-mounted audible alarm will be provided that activates whenever any fault status indicator or the "SWITCH POSN" indicator is lit, except when silenced intentionally by actuation of the port or starboard system "fault/silent" alternate action switch.

g. Central Processor Unit (CPU) - The CPU will contain no controls or indicators outside enclosure, except illuminated power supply fuses. The CPU will be located in the ship's central control station, in close proximity to the Central Control Unit (CCU), and will operate from 115 volt 60 hertz shipboard electrical power. A block diagram of the CPU is provided as Figure A3-1.

h. Local Control Unit (LCU), Port or Starboard - A guidance configuration of the LCU is shown in Figure T2. The port and starboard local control units (LCU's) will be identical and will be installed on the port and starboard fin subsystems, respectively, or in close proximity thereto, and will be located so as to be readily accessible for operation or servicing.

i. Auxiliary Sensor Unit (ASU) - This unit will provide auxiliary roll angle and roll rate signals, and will be installed near the ship's axis of roll.

j. Angle Transmitter Unit (ATU), Port and Starboard - The port and starboard ATU's will be identical and will provide DC voltages proportions to the port and starboard fin angles, respectively. The port and starboard ATU's will be mounted on the port and starboard fin subassemblies, and will be mechanically coupled to the port and starboard fin stocks or tillers, respectively.

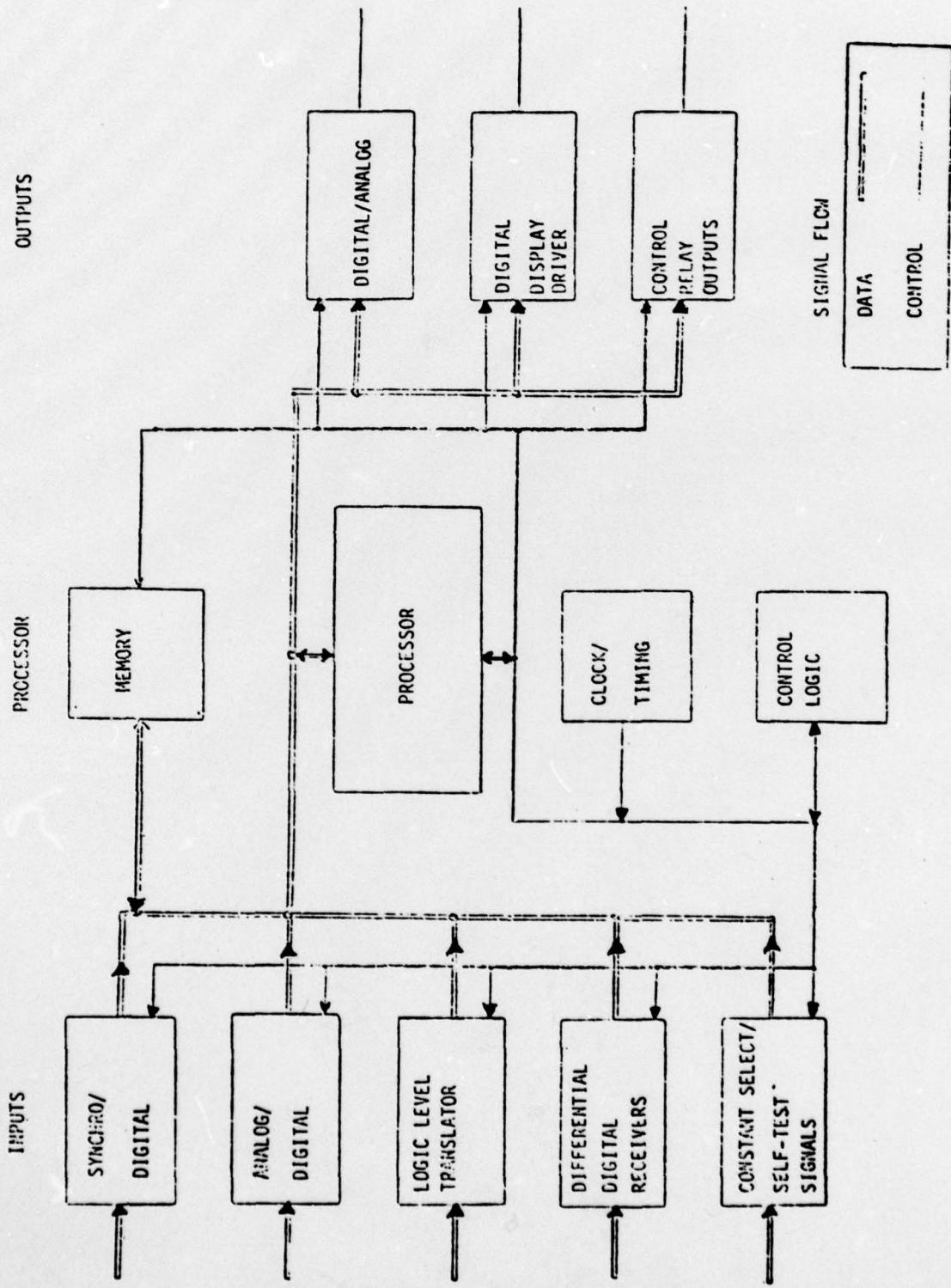


FIGURE A3-1. CENTRAL PROCESSOR UNIT BLOCK DIAGRAM